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CURRENT AFFAIRS

Notwithstanding the general impression that work in the engineering societies has been suspended for the summer, activities are nevertheless to be seen in many lines. On July 1 the usual number of certificates of indebtedness, amounting to $6000, were retired according to agreement, certificates being drawn by lot by the Chairman of the Finance Committee. This will be done annually until the entire amount has been paid, unless the matter may be financed earlier through the enterprise of the members and thus permit the greater activities of the Society, which are meanwhile diminished by this necessity.

ENGINEERING CONGRESS IN 1915

At a meeting of the representatives of the four national engineering societies, the American Society of Civil Engineers, the American Institute of Mining Engineers, the American Society of Mechanical Engineers and the American Institute of Electrical Engineers, on June 11, it was tentatively agreed to recommend to these organizations and to the Society of Naval Architects and Marine Engineers that they, together with the members situated on the Pacific Coast, underwrite the expenses of the engineering congress projected for 1915. Prof. W. F. Durand

of Leland Stanford University, and George W. Dickie of San Francisco also attended the meeting as guests.

SUB-COMMITTEES ON THE INDUSTRIES

A joint meeting of all the sub-committees on the industries and of the Committee on Meetings, was called for Monday, July 29, to formulate the scope of the work. These sub-committees, as already announced, include Iron and Steel, Air Machinery, Railroad Equipment, Industrial Building, Hoisting and Conveying, and Fire Protection, and it is expected that there will be professional sessions upon several or all of these topics at the Annual Meeting in December, with papers and reports on their various phases.

COMMITTEE ON TOLERANCES IN SCREW THREAD FITS

The interest aroused in the subject of the measurement of taps, bolts and screws at the meeting of the Society in New York in March, has been crystallized by the appointment by the Council of a committee on permissible Tolerances in Screw Thread Fits, L. D. Burlingame, Chairman, Elwood Burdsall, F. G. Coburn, F. H. Colvin, A. A. Fuller, James Hartness, H. M. Leland, W. R. Porter and F. O. Wells, which will investigate this important matter and report to the Council at an early date.

VISIT OF WILLIAM CAWTHORNE UNWIN

The Society anticipates with much pleasure the visit of William Cawthorne Unwin to this country about September 1. Dr. Unwin, who was elected to Honorary Membership in the Society in 1898, and who is also an Honorary Member of the Institution of Mechanical Engineers of Great Britain, was formerly professor of engineering of the Royal School of Naval Architecture and Marine Engineering, of the Royal Engineering College, and of the City and Guilds Central Technical College, and is the author of a number of books on hydraulics and machine design.

Dr. Unwin is President this year of the Institution of Civil Engineers of Great Britain.

EIGHTIETH BIRTHDAY OF JOHN E. SWEET

In honor of the eightieth birthday of Prof. John E. Sweet, Honorary Member and Past-President of the Society, it is planned to tender a reception to Professor Sweet about the time of the Annual Meeting, and a committee of arrangements, consisting of Ambrose Swasey, Chairman, F. G. Tallman, Vice-

Chairman, F. A. Halsey, Treasurer, E. J. Armstrong, Secretary, E. N. Trump, and Calvin W. Rice, Secretary of the Society, ex-officio, is preparing for such an occasion. On July 8 this committee met in Syracuse, N. Y., and made tentative plans for an event which would suitably express the personal affection held by every friend for Professor Sweet, to whom more than to any other living member is due the organization of the Society. It is expected that this anniversary will be one of the most delightful occasions of the mid-winter meeting.

CALVIN W. RICE, *Secretary*

MEDAL AWARDED TO GEORGE WESTINGHOUSE

George Westinghouse, Honorary Member and Past-President of the Society, was awarded the Edison Medal for "meritorious achievement in the development of the alternating-current system for light and power" at the annual banquet of the American Institute of Electrical Engineers in Boston on June 27. This medal, which was founded upon the initiative of the Edison Medal Association, comprising old associates of Thomas A. Edison, who subscribed a trust fund in 1901, is annually awarded by the Institute to that person in the United States or Canada performing the most meritorious achievement in electrical science, engineering, or electrical arts. It was designed by James Earle Frazer and bears on the obverse side a portrait of Thomas A. Edison, and on the reverse side an allegorical conception, the genius of electricity crowned by fame. The first award was made in 1909 to Elihu Thompson and the second in 1910 to Frank J. Sprague, the award to Mr. Westinghouse being the third. In accepting the medal Mr. Westinghouse gave an interesting sketch of the development of his inventions.

MEETING IN GERMANY

Plans for the meeting of the Society with the Verein deutscher Ingenieure in Leipzig, June 23-25, 1913, are already under way. The committee of arrangements appointed by the Council, consisting of Col. E. D. Meier, chairman, John W. Lieb, Jr., vice-chairman, Dr. W. F. M. Goss, Charles Whiting Baker, Major Wm. H. Wiley, and Dr. Alex. C. Humphreys, President, and Calvin W. Rice, Secretary, ex-officio, have opened negotiations with several steamship companies, and report that very reasonable figures can be obtained, despite the fact that the meeting is to be held during the height of the season, provided a sufficient number of members participate. By chartering an entire steamer demands varying from the most luxurious to the simplest and most economical can be adequately met. An early expression from members of the Society with regard to their intention to participate is greatly needed by the committee in order that further preparations may be made at once.

Letters received indicate the great interest felt by German engineers, manufacturers and officials in the proposed visit and more definite arrangements and further invitations from many cities and plants are contingent only upon the number and composition of our party. Besides Leipzig where the joint meeting takes place, many other points are on the alert to give the American engineers an opportunity to see the advances made in the profession and the growth of cordiality toward Americans generally, which is not confined to any one point. Places which it is tentatively proposed to visit are Hamburg, where there will be an opportunity to visit the greatest shipyards in Germany, Munich, with a reception by the Prince of Bavaria, Berlin, visiting the industrial plants, and by special arrangement the Castle of the Kaiser at Potsdam, Essen where there will be a reception by Krupp von Bohlen, Stuttgart, Düsseldorf, Cologne, Frankfort, Mannheim, and Heidelberg. In Leipzig it is also expected that there will be held during our visit a patriotic gathering in commemoration of the Battle of Leipzig, and it is expected that if the Kaiser is in the country at the time our party will be given a reception.

THE 1912 U. S. STANDARD

The American Society of Mechanical Engineers is a large factor in promoting the industrial progress of our country. This is usually accomplished by public spirited members serving on committees appointed for special tasks. The most recent development, recommended jointly by the Society and the National Association of Master Steam and Hot Water Fitters, is the Schedule of Standard Weight and Extra Heavy Flanged Fittings and Flanges known as "The 1912 U. S. Standard.". This work has been performed on request and the changes have been made for engineering reasons. They refer particularly to heavy work and to the sizes above nine inches. The above being true, these changes would affect principally the large consumers of great responsibility where safety is essential and where no expense would be considered too great which would prevent accident and secure continuity of service.

The various makes of flanges and fittings now on the market, figured on the basis of the bolt stress, show a factor of safety in many cases as low as two and in a few cases even less. This compared with the necessary factor of safety of the pipe itself, which is approximately sixteen, shows at a glance the necessity for the changes made in The 1912 U. S. Standard, of bringing up the factor of safety in the bolts, flanges and fittings, especially in view of the enormous strains encountered due to changes of temperature.

The other important feature in The 1912 U. S. Standard is the uniformity of dimensions providing for interchangeability of fittings. Until very recently, each manufacturer had his own standard and chaos has resulted, so that the engineer must know what particular make is to be purchased before he can proceed to design or erect the piping work. Hereafter all fittings of all manufacturers who comply with the new schedule will be uniform in the principal dimensions, and this will be an important factor in obtaining quickly spare parts throughout the country and enable one readily to make repairs and alterations. The new standard does not attempt to fix the quality of the metal or the thickness of the shell of fittings, leaving this to

the manufacturers to regulate in accordance with their guarantee, depending on the pressure the fittings are to carry.

STANDARD WEIGHT FLANGES

In comparing The 1912 U. S. Schedule of Standard Weight Flanges with the alternative standard submitted by the sub-committee of manufacturers with respect to diameter of bolt circle, number of bolts and diameter of bolt holes, we find they are identical, with the one exception that The U. S. Standard gives $\frac{7}{8}$ in. instead of $\frac{3}{4}$ in. for diameter of bolt hole for 4-in. pipe.

The standard of the Engineering Standards Committee of Great Britain and the standard of the Verein deutscher Ingenieure do not have sizes exactly corresponding to the standard weight and extra heavy weight of The 1912 U. S. Standard, but have the most important dimensions, the same for all pressures, making their piping on low pressures unnecessarily heavy and expensive, but without securing absolute safety for high pressures. Comparing the proposed Manufacturers' Standard with (Table II) British standard pipe flanges for working steam pressures of 225 lb. per sq. in., we find the British standards give higher values, especially as far as number of bolts is concerned.

EXTRA HEAVY FLANGES

For extra heavy flanges up to 9 in. inclusive, The 1912 U. S Standard and the standard of most manufacturers are identical. Above 9 in. The U. S. Standard is somewhat larger than the present standards, as has been stated. With the British standard, the diameter of bolt circle and number of bolts are the same for pipes corresponding to The 1912 U. S. Standard or proposed Manufacturers' Standard weight and extra heavy weight, but from 10 in. on the diameter of the bolt circle is from $\frac{1}{4}$ to $\frac{1}{2}$ in. less than the proposed Manufacturers' Standard and $\frac{3}{4}$ in. less than The U. S. Standard. The number of bolts in most cases for pipes from 10 in. on in The U. S. 1912 and Manufacturers' is from two to four bolts greater than in the British standard. As far as size of bolts is concerned, the British standard makes a distinction between pipes for pressures up to 225 lb. and up to 325 lb. The bolts for pipes up to 225 lb. are below the Manufacturers' Standard $\frac{1}{8}$ in. in nearly every case and below the U. S. Standard from $\frac{1}{8}$ to $\frac{3}{8}$ in. In pipes for pressure of 325 lb. the British standard is the same as the proposed Manufacturers' schedule in all cases except 14 in. and 18 in. where

the British standard is $\frac{1}{8}$ in. larger than the Manufacturers' standard, but from $\frac{1}{8}$ to $\frac{1}{4}$ in. smaller than the U. S. standard.

It is thus seen that wherever there is a distinction between The 1912 U. S. and proposed Manufacturers' Standard, The 1912 U. S. Standard is invariably on the side of greater safety and strength and has the added feature of interchangeability where the proposed Manufacturers' schedule has different face to face dimensions. As to the comparison with the British and German standards it must be remembered that they cover by one standard both weights of the American specifications, and as would be naturally expected with compromises must have higher values for standard weight pipes and lower values for extra heavy weight pipes. However, in Germany there is reported in the public press dissatisfaction with the existing standards as not being safe. The Alsace Association of Owners of Steam Machinery prints in its annual report for 1911 as follows:

"It is nearly impossible to estimate with any degree of precision the bending stresses to which are submitted pipe flanges when, in order to obtain a good joint, the bolts are screwed tightly, especially if the flange has to withstand at the same time stresses due to expansion. As a result, many of the joints in pipes and similar apparatus are working with very low factors of safety, as has been shown lately by the fact that accidents from piping have been far more numerous than from boilers, and with as fatal consequences."

The 1912 U. S. Standard is the result of the efforts of the aforementioned committees covering a period of over a year during which time every detail was considered. The committee had the active coöperation of engineers representing large power plants, manufacturers and the U. S. Navy Department. Prominent engineers and architects throughout the country were communicated with and their opinions and advice received and considered, and the consummation is The 1912 U. S. Standard.

This standard has been in operation two months and has already been approved and adopted by the U. S. Navy Department, U. S. Bureau of Standards, Isthmian Canal Commission, and is under consideration by other departments of the U. S. Government. Large power concerns such as the Interboro Rapid Transit Company of New York, the New York Edison Company and others who in the past have had fittings made from their own patterns, because there was no standard, have agreed to adopt and use The 1912 U. S. Standard. Mr. L. B. Stillwell, electrical engineer of New York, has specified The 1912 U. S. Standard for fittings in the new

120,000 kilowatt power house now in process of construction at Hauto, Pa., by the Lehigh Navigation Electric Company. We are informed that manufacturers are beginning to fill orders and in some cases are preparing the dimensions in their own new catalogues in accordance with The 1912 U. S. Standard. The committee of The American Society of Mechanical Engineers have decided after the most careful consideration not to reopen the subject, as has been erroneously reported.

This schedule was published in the February 1912 Journal of The American Society of Mechanical Engineers and February 1912 Bulletin of the National Association of Master Steam and Hot Water Fitters, and copies may be had upon request from either the National Association of Master Steam and Hot Water Fitters or The American Society of Mechanical Engineers.

NECROLOGY

GEORGE R. BABBITT

George Rodney Babbitt, president and treasurer of the American Oil Company of Providence, R. I., died at his home in Edgewood, R. I., on June 29, 1912. Mr. Babbitt, who was born at Berkley, Bristol Co., Mass., on June 27, 1842, received his education in the schools of that locality and in Fall River, Mass. As a young man he followed the sea for several years, making a voyage to California in 1860, then a tedious undertaking.

In 1862 Mr. Babbitt came to Providence and entered the employ of the Burnside Rifle Works, becoming a sub-contractor on parts of the work. When the works closed down three years later to be refitted for building locomotives, he spent a period in the Pennsylvania oil fields, and the following year became associated with William A. Harris, the Harris-Corliss steam engine builder. With this concern he filled the positions of machinist, foreman and superintendent, and invented and patented several improvements, among them a piston packing, for which he received a medal at the Cincinnati Industrial Exposition in 1874, a releasing gear and an extended piston.

In October 1888, he was tendered the position of superintendent of the heavy ordnance department of the Government at Washington, D. C., which, however, he declined. He bought an interest in the American Oil Company of Providence in 1892, with which he was associated at the time of his death. He was frequently called upon to act as a consulting and mechanical engineer for large manufacturing concerns, both for new work and for the best methods of repairing. In 1894 he patented an automatic barrel filler.

One notable achievement of Mr. Babbitt's was his plan for floating the steamship Paris which had grounded on some rocks while crossing the Atlantic, and which he proposed to the owners after reading an illustrated newspaper description of the accident.

WILLIAM J. EDWARDS

William J. Edwards died at his home in Binghamton, N. Y., on April 21, 1912. Mr. Edwards, who was born in Albany on February 20, 1873, received his preparatory education at the Centenary Collegiate Institute at Hackettstown, N. J., and his technical training at Pratt Institute in Brooklyn. Upon his graduation in 1893 he entered the employ of the Tide Water Oil Company in Bayonne, N. J., where he became acting master mechanic of the company, with charge of the design and construction of the complete oil refinery, and later superintendent of the can and case department. He was also during his residence in Bayonne president of the Bayonne Castings Company, which made a specialty of monel metal castings.

In 1911 he removed to Binghamton, N. Y., to become vice-president and treasurer of the Binghamton Clothing Company, a position which he held at the time of his death.

LAWRENCE WHITCOMB

Lawrence Whitcomb, who died in Boston, Mass., on May 18, 1912, was born May 2, 1863, at Malden, Mass., and was educated in the Boston and Roxbury public schools. In 1881 he left school without completing his course and entered the shoe business, becoming a salesman and factory superintendent. In 1890 he joined the organization of the firm of Whitcomb, Weard & Company, real estate brokers and agents, which became one of the large real estate firms of Boston. Mr. Whitcomb was also interested in the development of the National Brake & Clutch Company and its successor the Cork Insert Company, which owned patents covering the use of cork for friction purposes in brake shoes, clutches, pulleys, etc., and finding that the use for these devices was very large he made a careful study of mechanical conditions and requirements and patented several new applications of his own. He was at the time of his death treasurer and general manager of this company.

THE REDUCTION IN TEMPERATURE OF CONDENSING WATER RESERVOIRS DUE TO COOLING EFFECTS OF AIR AND EVAPORATION

By W. B. Ruggles, Published in The Journal for April

ABSTRACT OF PAPER

The tests described in this paper were made to determine the heat radiation from a cooling reservoir of about 6½ acres, at the Crescent Portland Cement Company's plant at Wampum, Pa. The amount of heat delivered to the reservoir from the engines and compressors was measured by hourly readings for power and vacuum and the reduction of temperature, due to pumping in fresh water from the Beaver River, and by rain was also taken. By frequent readings of temperature of tail water, intake water and air, the cooling effect of the air on a unit of surface of water per unit of difference of temperature was easily determined.

DISCUSSION

Forrest E. Cardullo. It may be shown from the kinetic theory of heat that the rate of evaporation from a water surface is approximately proportional to the difference between the saturation pressure of water vapor at the temperature of the water and the actual pressure of the water vapor in the air. Experiment shows that the rate of evaporation in still air, i. e., air in which the only currents are those caused by the temperature conditions and not by the wind, the rate of evaporation is approximately 3/10 lb. of water per sq. ft. of surface per hour for every pound difference in vapor pressure. When there is an appreciable amount of wind, the rate of evaporation becomes higher, and in case the wind is very brisk it may become four or five times as great as the value given. Furthermore, the rate of evaporation depends on the size and form of the pond, being

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The American Society of Mechanical Engineers, 29 West 39th Street, New York. All discussion is subject to revision.

greater in the case of a small and narrow pond and less in the case of a large or circular pond, since the air which sweeps across the large pond will be partially saturated with moisture before it has gone very far.[1]

There is a reasonable concordance between theory and the actual results which Mr. Ruggles obtains. For the week ending May 7 the amount of heat carried away was about 15 per cent greater than theory would indicate. For the week ending July 12 it was a few per cent less, but for the week ending November 27 it was nearly twice as great.

This is not contrary to theory as it may seem to be since one very essential element has been omitted in the data of Mr. Ruggles' paper, namely the velocity and direction of the wind, and it is reasonable to assume that in November the amount of wind would be considerably greater than during May or July. The direction of the wind would also have a considerable effect, since a wind blowing the long way of the reservoir would produce a lower rate of evaporation than one which blew crosswise.

Mr. Ruggles gives for the average temperature of the reservoir the mean of the temperature of the intake and the condenser discharge. As a matter of fact, it could be but little higher than the temperature of the intake. He assumes that the steam rejected by the engines was of 100 per cent quality. The quality was probably not far from 85 per cent and might be as low as 80 per cent. His figures on the amount of heat delivered to the reservoir are therefore about 20 per cent too high.

An inspection of Mr. Ruggles' paper would lead one to expect that a cooling pond would dispose of 3½ b.t.u. per sq. ft. per hr. per deg. difference in temperature between the water in the pond and the air. When, however, the temperature of the condenser intake is taken for the temperature of the pond, and a quality of 84 per cent is assumed for the steam discharged to the condensers, the b.t.u. per sq. ft. per hr. per deg. difference in temperature ranges from 4.13 to 7.60, being higher in hot weather than in cold weather. This is as it should be, since the relative humidity is less and the vapor pressure difference and rate of evaporation are greater in hot weather.

The average rate of evaporation for the week is not the rate

[1] For the development of this theory and its application to the design of cooling ponds see Practical Thermodynamics, McGraw-Hill Book Co., pp. 264-267.

of evaporation which would be produced by the average conditions of temperature and humidity for the week. On this account, and because the weather cannot be predicted in advance, and on account of the great variations of weather conditions to be expected, it is foolish to attempt any degree of refinement in the design of cooling ponds. It is not desired to design a cooling pond which will give an intake temperature of 90 deg. at 2 o'clock in the afternoon on July 4, but to design one which will, when used in connection with the other apparatus of the station, give the required power output at a minimum total cost. Either the data given by Mr. Ruggles or the method mentioned in this discussion will enable one to do this.

WM. T. DONNELLY said he had discussed the elements of the subject with the author before the design of the reservoir and afterwards had discussed the results. It seemed to him that more depended upon the humidity of the air than upon any other single factor. The atmosphere must be considered as a condenser and the rate of flow of vapor into the atmosphere is affected very much by the condensation.

In the winter the greater evaporation is due to the relatively lower humidity of the air, and the passing off of the vapor from the warm surface is more rapid on that account. The author is to be commended for giving information along new lines dealing with manufacturing conditions as met with all over the country, and it is not necessary to have it to the third or fourth decimal place to make it useful.

WILLIS H. CARRIER. As Professor Cardullo suggested, it would seem that the rate of heat transfer would be in direct proportion to the total heat difference in the air at the air temperature, and at a saturated air temperature, corresponding to the water temperature of the air in the pond. In other words, there is a certain amount of heat in the natural air to start with, which being in immediate contact with the pond is raised to and saturated at the temperature of the pond, so that the amount of heat taken away from the pond under the same conditions of wind would be in direct proportion to this total heat difference. This would naturally lead to the conclusion that there would be a much higher rate per degree difference at the high temperatures as the total heat of saturated air increases very rapidly at the

higher temperatures. But this is not so, and Professor Cardullo has offered an explanation why it is not. From an engineering standpoint it is very interesting to know that the rate of heat transfer is much greater in the months when the air is cooler and the temperature of the pond is lower, so that, as Professor Cardullo says, calculations may be made upon the summer conditions with very little wind.

There is also another point to be taken into account where there is no forced circulation, that is, that when the temperature increases, the rate of transmission would also be increased by the fact that the natural air currents produced by this temperature are increased, producing an induced circulation. Just what the effect of this is and what the laws governing it are is very difficult to determine, but probably they are very similar to those of direct radiation, which would give a greater rate of transfer at a greater temperature difference. This, however, does not seem to be of much interest from a practical standpoint. So many variables enter into the conditions that the approximate constants as shown by the results of these tests may be accepted as the average conditions.

Mr. Carrier believes that certain statements made by Mr. Donnelly with reference to the effect of the humidity upon the rate of heat transfer may be somewhat in error. It was stated that the greater heat transfer may be accounted for in cooler months by the lower relative humidity. The fact is that it is the reverse. The relative humidity according to all weather bureau readings is lower during the hotter months than during the spring and fall. There is a considerable higher relative, not absolute, humidity in the fall and spring. The question is rather one of wet-bulb temperature, which is the only temperature worth considering, because that is the measure of the total heat in the air, and it is the temperature at which water will arrive if subjected to the air for a sufficient time. The effect of the relative humidity, of the lower relative humidity, is one of a low wet-bulb temperature, and the amount of evaporation that occurs is measured by the difference between the water temperature and the wet-bulb temperature. The relative humidity plays but a small part in this.

THE AUTHOR. There is no doubt, as Professor Cardullo states, that the direction and force of the wind have considerable influence on the amount of evaporation, but as these tests were made

in order to obtain some practical and reasonable basis for estimating the size of a reservoir required for any particular plant and not for a mere academic discussion of theories, the amount of wind was purposely omitted as introducing an uncertain factor which would be of no future use. Moreover the author considered the average of three different weeks widely separated would give a fair average of yearly conditions.

Professor Cardullo is mistaken regarding the average temperature of the pond being nearly the temperature of the intake, and the author cannot understand where he could get such information. Actual readings 100 ft. apart show the mean temperature, as he stated, to be the average between the tail water temperature and that of the intake water.

Presumably Professor Cardullo in referring to the quality of the steam being only about 80 per cent or 85 per cent meant the percentage of dryness and probably had in mind the steam as it left the low-pressure cylinders or as it entered the condenser. But as the amount of steam figured was that which entered the engine and was corrected by calorimeter test for moisture, and as all the heat, except that utilized for power, went into the condenser and reservoir and a very small percentage was due to radiation from the engine cylinders, his criticism of the tests in this respect is futile.

In making these tests the author had no idea that the results obtained would be absolute, but thought that they would throw light on a subject which as far as he was able to learn had not been treated before.

RESULTS OF TESTS ON THE DISCHARGE CAPACITY OF SAFETY VALVES

By E. F. Miller and A. B. Carhart, Published in The Journal for February

ABSTRACT OF PAPER

The paper gives the results of a series of tests made on the discharge capacity of 3-in. and 3½-in. locomotive pop safety valves at varying lifts under 200 lb. boiler pressure, also results of tests on the discharge capacity of 3-in. and 3½-in. inspector valves of the flat-seated and bevel-seated types under 100 and 150 lb. gage pressure. In all of the tests the steam was condensed in a surface condenser and the condensation weighed. The tests were run from 30 minutes to 1 hour in duration so that errors of weighing were reduced to a minimum. In all of the tests the valve was set a definite distance from its seat and held rigidly, the spring of the valve having been removed.

Results of the tests are given both in the form of a plot and numerically.

DISCUSSION

Chas. H. Chase. The records of these tests give an opportunity for the calculation of an average constant for comparison with the constant of Napier's equation and with the constant determined by a series of experiments which were described in a paper [1] presented before the Society in 1909.

That series of tests was made on 3-in., 3½-in., and 4-in. valves of the stationary type, and on 1½-in., 3-in. and 3½-in. locomotive type valves. The valve springs were removed and the lift of the valve disc was controlled in each test by a solid threaded spindle carried by the valve case. The discharge was found by weighing the feedwater, the water level being kept constant. The steam had an average superheat of 37.2 deg. fahr. All of the valves had 45 deg. bevel seats, and the discharge areas were found by the equation

$$a = 2.22 \times D \times l + 1.11 \times l^2$$

except in two of the 24 tests where the valve lift was greater than

[1] Safety Valve Capacity, P. G. Darling, Trans. Am. Soc. M. E., vol. 31, p. 109.

Presented at the Boston Meeting, March 16, and at the Spring Meeting, Cleveland, O., 1912, of The American Society of Mechanical Engineers.

the depth of seat and then the equation used was

$$a = 2.22 \times D \times d + 1.11 \times d^2 + D \times (l - d)$$

in which D = diameter of valve, l = valve lift, d = depth of valve seat.

The average constant for the tests was 47.5, to compare with

TABLE 4 VALUES OF CONSTANTS WITH LIFTS AND AREAS OF DISCHARGE

3-In. Regular Bevel Seat
Pressure 100 Lb. Gage

Lift	E	=	Constant	×	a	×	P
0.0207	806	=	50.8	×	0.1383	×	114.7
0.0507	2054	=	52.6	×	0.3405	×	114.7
0.0807	3264	=	52.1	×	0.5447	×	114.7
0.1007	4065	=	52.0	×	0.6819	×	114.7

3-In. Regular Bevel Seat
Pressure 150 Lb. Gage

Lift	E	=	Constant	×	a	×	P
0.021	1225	=	53.0	×	0.1403	×	164.7
0.051	2961	=	52.5	×	0.3425	×	164.7
0.081	4608	=	51.2	×	0.5467	×	164.7
0.1010	5743	=	51.0	×	0.6840	×	164.7

3½-In. Bevel Seat
Pressure 100 Lb. Gage

Lift	E	=	Constant	×	a	×	P
0.0215	972	=	50.6	×	0.1676	×	114.7
0.0515	2384	=	51.5	×	0.4031	×	114.7
0.0815	3810	=	51.9	×	0.6406	×	114.7
0.1015	4690	=	51.2	×	0.8001	×	114.7

3½-In. Bevel Seat
Pressure 150 Lb. Gage

Lift	E	=	Constant	×	a	×	P
0.0222	1484	=	52.1	×	0.1730	×	164.7
0.0522	3495	=	51.9	×	0.4086	×	164.7
0.0822	5454	=	51.2	×	0.6462	×	164.7
0.1022	6854	=	51.6	×	0.8057	×	164.7

the value 51.4 in Napier's equation

$$E = \text{constant} \times a \times P$$

for discharge in pounds per hour.

In the tests of 3-in. and 3½-in. valves with bevel seats now under discussion, the lift was greater than the depth of valve seat at 0.08 and 0.10 in. However, as a perpendicular from the lower angle

of the valve disc does not clear the seat in either case, there remains a discharge orifice bounded by two parallel conical surfaces as on the lower lifts, and the same method of calculation for discharge area may be used.

Table 4 gives the values of the constants together with the lifts and areas of discharge. Here the average value of the constant is 51.7.

If the calculation of discharge areas for lifts greater than the depth of valve seat is made in the same way as in the tests quoted, the average value of all the constants becomes 49.0. That method is, however, inadmissible for the reason given in Par. 4 of this discussion and we are brought to the conclusion from these tests that for lifts up to 0.10 in. the discharge of dry steam for valves with

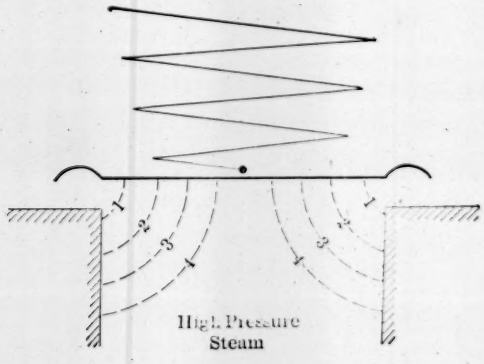

FIG. 16. SHOWING VARYING PRESSURES AFFECTING LIFT OF VALVE

bevel seats may be found very closely by the direct application of Napier's equation.

W. TRINKS. It occurs to me that the most important part of Professor Miller's paper has been omitted. We can predict with some degree of accuracy how much steam a valve is going to discharge for a given lift, but it takes a very wise man to predict the lift, for this reason: The region around the valve seat is a nozzle so that at line 1, Fig. 16, the pressure will be less than at line 2 and that again less than at line 3, etc.; but the pressure over the whole plate, plus the lip for popping, keeps the valve open against the compressed spring. For heavy flow there must be equilibrium between spring force and static and dynamic steam forces. To pre-

dict the lift of the valve one needs the distribution of the pressure lines 1, 2, 3, 4, etc., over the valve face. While the tests in question were made the distribution of pressure might have been tested. It is desired to know whether such tests were ever made and if so, when they have been published.

A. B. CARHART. The tests were part of the experimental work incidental to proposed improvements in safety valve design, and the measurements obtained were intended to be kept as private information. The purpose was not to determine the maximum amount of steam which could be discharged under any extreme conditions of valve adjustment. The tests give simply the desired positive and quantitative measure of the effect of certain definite modifications of the contour of the orifice at the valve seat, in forms adapted to ordinary commercial uses.

Special care was taken to insure dry steam, and this was shown to be at all times between 0.995 and 0.999 dry, measured close to the safety valve inlet. Obviously the weight of steam apparently passed would have been considerably greater if it had been wet. The steam was measured by condensation of the discharge, for weighing the feedwater would have given only approximations that would not be tolerated in boiler tests or other steam measurements.

The form of the curves, which closely approximate straight lines passing through the origin at zero, indicates well designed proportions of the orifices at the various lifts, and tends to the conclusion that the curves may reasonably be projected further to indicate the discharges for the slightly higher lifts sometimes desired and attained in these valves under actual working conditions.

As the basis of comparison with the results that may be predicted by applying Napier's formula to the calculation of safety valve discharges, the constant derived from Professor Miller's tests might perhaps be more conveniently taken with reference to 70, as used in the ordinary form of Napier's equation, rather than with the transposed value of 51.4 referred to in the form used by Professor Chase. The discharges measured by Professor Miller show a value of the constant of 69.68 as the average of all his measurements of the bevel-seated valves, or a value of 69.93 as the average of the eight tests at the normal lifts of 0.08 and 0.10 in., showing a very close approximation to the value of 70 of Napier's formula. This seems to establish the reasonableness of applying Napier's equation to

the calculation of the discharge of safety valves of the type tested by Professor Miller, but the writer thinks these results will hardly justify the general conclusion stated by Professor Chase that "the discharge of dry steam for valves with bevel seats may be found very closely by the direct application of Napier's equation," for obviously the equation can hardly be directly applied to all valves, since in the earlier tests referred to by Professor Chase, with which he makes comparison, the discharges were apparently only about 92½ per cent of the values given by Napier's formula, as Professor Chase shows.

In comparing the present bevel-seated valves with other valves, Professor Chase disregards the fact that although both are called "bevel-seated" (having the contacting faces at an angle of 45 deg. to the vertical), yet the form of the approach to the seat orifice, and the passages and chambers at the lip of the valve were entirely different in design, the extreme lip of the disk of the valves in the earlier tests turning upward, instead of downward as in the valves tested by Professor Miller, and in other ways differing so that they can hardly be directly compared. Moreover, in the earlier tests, no account was taken of the difference in the expansion of the spindle and the valve body, or of the additional lift due to the pressure of the steam against the disk. Professor Miller shows that these may be considerable in amount. The bevel-seated valves tested by Professor Miller were held rigidly in place by screw threads directly below the seat, ending practically on the level with the steam-tight joint, yet even in this case the additional strain beyond the set opening in the various cases was 0.0007, 0.0010, 0.0015 and 0.0022 in., respectively, which gave an additional area of opening, beyond the amount due to the nominal measured lift, averaging 2.9 per cent and in the maximum case amounting to 7.5 per cent increase. In the case of the flat-seated valves, where the control was about 2 in. above the valve seat (Fig. 2) the additional strain due to the steam pressure beyond the measured lift was in the various cases 0.0024, 0.0044, 0.0022 and 0.0032 in. This is several times greater than in the case of the bevel-seated valves, which were controlled much nearer the valve seat; and this greater percentage of increase in discharge area was found, although the area of the disk upon which the additional pressure was exerted is much smaller than in other valves. If the screw-thread control of the spindle carrying the disk were still further from the

seat, at a distance of perhaps 8 or 10 in., the amount of actual opening of the valve beyond the measured opening as set must inevitably be considerably greater than the amounts determined by Professor Miller, and would thus modify the calculation of the percentage of discharge and the relative value of the constant to be compared with Napier's constant in such cases, so as to have reduced appreciably the apparent indicated percentage efficiency of discharge. In Professor Chase's illustration of the effect of using an erroneous formula, which he says is inadmissible, he applies it to 12 cases out of 24 in Professor Miller's report; whereas in the earlier tests he refers to, it was used in only two cases out of 24. Therefore the value of the constant as then determined could not have been greatly affected.

If it were to be assumed, in the case of the flat-seated locomotive valves tested by Professor Miller, that all of the steam discharged had passed over the outer seat alone, the percentage of the discharge would have averaged 108.5 per cent for the valves having the square edge seat and 121.2 per cent for the valves with the seat having the rounded edge, as compared with the discharge to be predicted for the same areas of discharge orifice, calculated according to Napier's formula, taken as 100 per cent. This simply establishes what has been so often asserted, that approximately 15 to 18 per cent of the total discharge of these valves is through the openings through the hollow arms leading from the central well, by which the amount of the valve lift and the closing pressure are regulated, affording therefore this additional by-passed flow of steam through the central passages, independent of and in addition to the main discharge over the outer flat seat. And there is the other interesting comparison, that by reason of these seats being flat, instead of beveled at an angle of 45 deg., the calculated discharge orifice, for the same lift, is nearly 1½ times as great as in the valves of the same seat diameter with the 45 deg. angle.

The original and only purpose of Professor Miller's tests of the locomotive valves was to determine the actual increase in the rate of steam discharge due to rounding the edge of the valve seat. In order to measure exactly the effect due entirely to this change in design, it was necessary to remove the springs and to secure the valve disks at the predetermined lifts. When such valves are operating in ordinary service with the usual springs the amount of lift is undoubtedly not always the same and is not

uniformly maintained throughout the period of blowing: nevertheless these tests do establish the fact that if these valves are regulated under the operation of the spring so as to lift 0.05, 0.08 or 0.10 in., as may be desired, then the steam discharge must

TABLE 5 BEVEL-SEATED, IRON-BODY VALVES AT 100 LB. AND 150 LB. WORKING PRESSURE. AS TESTED BY PROFESSOR MILLER

Size of Valve, In.	Actual Lift	Area of Orifice at Seat Opening	Average Boiler Pressure	Discharge in Lb. per Hour Measured by Tests	Discharge in Lb. per Hour Predicted by Napier's Formula for Given Area and Pressure	Discharge, Percentage, Efficiency upon Napier's Predicted Discharge	Equivalent Constant —Compared with Napier's Value 70
3	0.0207	0.1384	110.4	879	890.43	98.72	70.92
3	0.0507	0.3406	109.2	2219	2170.30	102.24	68.49
3	0.0807	0.5447	108.3	3500	3445.61	101.58	68.96
3	0.1007	0.6820	109.8	4412	4366.75	101.04	69.44
3	0.0210	0.1404	161.9	1313	1275.15	102.97	68.02
3	0.0510	0.3426	154.9	3049	2988.26	102.03	68.49
3	0.0810	0.5468	152.2	4670	4693.42	99.50	70.42
3	0.1010	0.6840	151.3	5788	5839.40	99.12	70.42
3½	0.0215	0.1676	110.2	1058	1076.57	98.28	71.42
3½	0.0515	0.4031	105.0	2488	2481.48	100.26	69.93
3½	0.0815	0.6407	102.0	3878	3845.30	100.85	69.44
3½	0.1015	0.8001	102.4	4788	4818.43	99.37	70.42
3½	0.0222	0.1730	152.2	1504	1484.93	101.28	68.96
3½	0.0522	0.4086	151.1	3518	3484.07	100.97	69.44
3½	0.0822	0.6462	154.9	5616	5636.34	99.64	70.42
3½	0.1022	0.8057	147.4	6746	6716.77	100.43	69.93

Average of 16 tests, percentage efficiency 100.517
Average of 8 tests, at normal lifts of 0.08 and 0.10 only 100.191
Average of 16 tests, equivalent value of constant 69.68
Average of 8 tests, at normal lifts of 0.08 and 0.10 only 69.93

Orifice areas calculated by full formula, $A = 2.22\ Dl + 1.11\ l^2$.

Data taken from Professor Miller's test of 1910.

Showing rates of discharge slightly exceeding rate predicted by Napier's formula, but fairly confirming its accuracy, as applied to these values.

Napier's formula, W (per second) $= \text{Area} \times \frac{\text{Abs. Press.}}{70}$

inevitably be at the rates given in this report. He showed for example that the valves with the rounded seat edge discharged an average of 11.5 per cent more steam than the valves with the square edge seat, the increase being 12.3 per cent in the case of the 3-in. locomotive valve at 0.05 and 0.10 in. lift.

As it was not Professor Miller's purpose in measuring the discharges of the bevel-seated and flat-seated iron-body valves, to make any direct comparisons with the results of the tests of any other valves or with any other experiments made at different times for other purposes, the writer has not attempted to present any such calculations. But the actual measurements showed an average discharge by the flat-seated iron-body valves 46.5 per cent greater than in the bevel-seated valves at the same fixed lifts; the increase, for example, averaged 52.2 per cent in the case of the 3-in. valves at 100 lb. pressure. The discharge of all the 3-in. flat-seated valves was 49.5 per cent greater than of all the 3-in. bevel-seated valves, and the average increase was 49.2 per cent in the case of all flat-seated valves of both sizes at 100 lb. pressure.

Table 5, showing the comparison of Professor Miller's results with the discharges to be predicted by applying Napier's formula to the nominal discharge areas through the seats of the bevel-seated valves, may be of interest.

E. F. Miller desires to present no closure.—EDITOR.

THE PRESENT STATE OF DEVELOPMENT OF LARGE STEAM TURBINES

By A. G. Christie, Published in The Journal for May

ABSTRACT OF PAPER

This paper deals with the present state of development of leading types of large steam turbines, some details of construction, the commercial results obtained and some new uses to which steam turbines have been put. It also points out the probable tendencies of steam-turbine development.

For the purposes of discussion, large steam turbines are divided into two types, fundamental and modified or combined. The weak elements of the fundamental types are discussed and the advantages of the new types pointed out. Some new constructons are shown in section.

The details of construction are discussed fully. A table with brief notes on the details of construction of individual types is included for rapid comparison of the practice of various manufacturers. The different types are also compared on the basis of efficiency as shown by published results of tests.

The present status of low-pressure turbines, turbo-compressors, turbo-driven pumps, geared turbines and marine turbines is discussed briefly. The closing paragraphs deal with the probable trend of steam turbine development and future possibilities.

DISCUSSION

F. Hodgkinson. In the table showing the characteristics of turbines of different manufacturers there are certain points that the writer would like to add to or correct in reference to the machines built by the Westinghouse Machine Company, first respecting speeds.

Speeds have been increasing lately for 60-cycle work. For sizes of from 3000 to 10,000 kw. 1800 r.p.m. is the speed employed and for larger powers, 1200 r.p.m. For 25-cycle work, 1500 r.p.m. is employed up to a rating of 15,000 kw. (normal), or 20,000 kw. (maximum). A speed of 750 r.p.m. is not likely to be employed for capacities of less than 30,000 or 40,000 kw.

The American Society of Mechanical Engineers, 29 West 39th Street, New York. All discussion is subject to revision.

The speed of impulse blading, as employed by the Westinghouse Company, varies with the different types from 350 to 500 ft. per sec. The speed of the reaction or Parsons blading at the low pressure depends upon the steam volumes involved and varies from 300 to 500 ft. per sec., the latter in the case of large machines where constriction in the exhaust must be avoided on account of the large volume.

The journal speeds are higher than Professor Christie points out. Journal speeds reach 80 ft. per sec. and pressures 100 lb. per sq. in. of projected area, and this pressure may be materially increased without encountering wear of the bearings, provided the proper flooded lubrication is employed and bearings are sensibly designed. With higher surface speeds a little more oil-cooling capacity will be necessary on account of the greater rate of shear of the oil. A good warm temperature, not less than 150 deg., is conducive to high economy.

Concerning the economies of turbines, Professor Christie quotes the opinion of European engineers, that higher economy is to be obtained from impulse elements of the Rateau or Zoelly types, rather than with impulse elements of the Curtis design. The writer believes this to be a generally accepted fact and in the case of the latter type is doubtless due to the low efficiency of the last row of moving blades. Apparently having once expanded, steam eddies are brought about each successive time the steam is put through the blade elements. Somebody has remarked that the fourth row in a Curtis element has never been found useful. Theoretically it is useful, but in practice, except for very high pressure drops, it has not been found so.

It would seem that the reaction turbine, unless it be subjected to some disability, may be more economical than either of the other types. With any impulse turbine there are necessarily two complete conversions of energy, the first expanding the steam through the nozzle, the steam doing work in giving itself velocity; and the second extracting the velocity of the steam. Both of these transformations of energy are subject to losses, the second greater than the first. If the nozzles were permitted to revolve, and the turbine had no blades at all, obviously such a turbine would be the more efficient. The reaction or Parsons turbine is such a machine, but, unfortunately, it is subject to leakage. However, with higher speed machines and capacities that lend themselves to good proportions of blading, this leakage, particularly

at the low-pressure end, is a very small factor, and such a turbine is the more efficient. These features are emphasized in the later high-speed turbines, particularly of the combination type, comprising a Curtis turbine element for the high pressure, and a reaction element for the low pressure, the Curtis element as referred to the regular Parsons, replacing a large number of rows of reaction blading at a small diameter with the attendant greater axial length, or a smaller number of rows of larger diameter, the blades being smaller in the latter case, and of poor proportions, subject to high leakage.

Of late, the tendency has been toward higher speeds, which, within reasonable limits, is conducive to better results, not only from the standpoint of economy, but also from the standpoint of operativeness. As Professor Christie has stated, congestion at the low-pressure end may be entirely avoided by making the last portion double-flow. Higher rotational speed involves smaller diameters, and hence better blade proportions at the high pressure end, with the attendant lesser leakage. In addition to this, the turbine cylinder structure as a whole is much smaller, there is less distortion of structure with temperature changes, thus permitting running the blades with smaller clearances if it is desired to do so.

Professor Christie very properly points out that there is a lower leakage ratio in a single-flow turbine than in one of double-flow construction. However, the choice of revolutions is limited on account of the standard frequencies employed. While reducing the speed to the next lower one available may permit of single-flow construction, the advantages may be entirely offset by the increased clearances which must be employed, simply because the size of the machine has so materially increased, with the attendant greater amount of distortion. Therefore, the turbine having the low-pressure sections double-flow is frequently the more economical machine. It is very convenient to be able to double up the steam passage at the low-pressure end, because of the tremendous rate with which the volume increases, particularly with machines designed for high vacuum.

Concerning blade troubles and blade construction, the writer ventures the statement that no turbine built by a reputable manufacturer in this country ever came to grief because of the blades becoming detached or breaking, due in itself to centrifugal force. Accidents have occurred because of misalignment or distortion,

resulting either in blades colliding with turbine parts, or with one another, which of necessity resulted in injury. The principal source of blade trouble is due to the blades buzzing or vibrating in the steam current. The natural vibrating period of the blade may be raised by means of a shroud, or other form of lashing, but not eliminated.

Fig. 12 shows what has now become the standard construction for reaction blading, as employed by the Westinghouse Machine Company. The feature is an interlocking system, the strength of which is equal to that of the blade itself. The blade sections are stronger at the root than at the tip. The section tapers

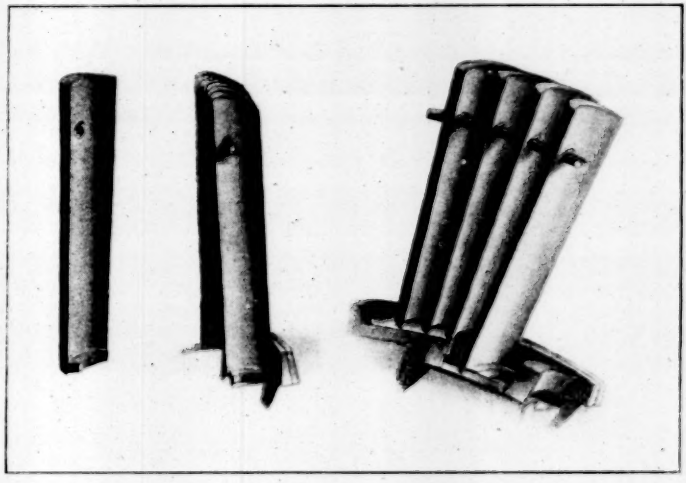

FIG. 12 BLADING USED IN WESTINGHOUSE TURBINES

slightly from the root, thus eliminating the chance of injury due to any vibrating or buzzing, should it exist.

As shown in Fig. 12, the bottom end of the blade is upset, which is truly an upsetting process, and not any mere bending, and the die for holding the blade at this time is so arranged that the section of the blade is increased in thickness throughout its lower portion, this latter feature, however, not being clearly shown in the illustration.

The dove-tailed groove in the blade carrying member is provided with a supplementary groove in the floor of the main groove, into which fits the upset end of the blade. The latter locks underneath the packing pieces which occupy the main

groove. The larger blades are provided with compound wedges which go between the walls of the groove and the blades and packing pieces, which effectively fills the groove in a more satisfactory manner than a calking strip. In the case of the smaller blades, the wedges are found to be unnecessary, owing to the lighter sections of the packing pieces, which permits them to fill the groove, when driven into place. Inasmuch as no calking of any kind is resorted to, blades may be taken out and replaced an indefinite number of times.

The author has brought out an important point in the matter of standardization of ratings for turbines, and suggests that the Society fix a rating for turbines, defining their overload capacity. While this may be a little difficult to do, the writer cordially concurs with him that a great deal of confusion exists.

In discussing capacities of machines, one is compelled to go to some explanation as to what is meant. The kilowatt rating needs as much explanation as does the load factor.

Present methods cause particular confusion in the statements of the cost per kilowatt of the installation of a turbine station. To standardize ratings may be difficult unless the turbine be dissociated from the generator. There are, however, two needs of ratings for power-plant work today. One is for the very large power plants, such as the Edison plants, which have a large number of units and large capacity, and the condition is such that if the machine is running at all it is running at its highest load. Obviously a generator in such a plant may advantageously be given a certain maximum continuous rating at a specified temperature rise, and the turbine connected to it arranged to give its highest economy at that load, and need have no overload capacity, or at most 10 or 15 per cent.

The other kind of rating required is that needed by the smaller plants with few units, the plant having to encounter heavy peaks at certain times of the day. Here, while the generator may have a certain maximum capacity as to its basis of rating, the turbine should have its highest economy at some fraction of this load. For such an application the old method of rating machines at a normal full load, with a 60-deg. rise, would seem to apply to the turbine having its best economy at that load and the generator capable of carrying 25 per cent overload continuously and both turbine and generator capable of running for a limited time at 50 per cent overload.

That system of rating was first practised in this country in the early days of turbine development and had not been practised in Europe. It was brought about by the turbine having to compete with the Corliss engines. Latterly, bargain-counter methods of buying (an expression borrowed from Professor Christie) have caused such outfits to have their turbines sometimes given a maximum rating corresponding to the 50 per cent overload just referred to. It would seem entirely feasible to have generators uniformly rated on maximum continuous load basis; the turbine, however, to have its best steam consumption at some fraction of this, depending on the requirements of the installation, and with overload capacity to suit the maximum possible load on the generator.

J. A. Moyer. Interest in Professor Christie's paper, in the writer's opinion, centers about his tabulation of recent tests of steam turbines. Such a tabulation is, however, scarcely complete if it does not include the very elaborately detailed and accurate test of a Sulzer turbine, of the combined impulse reaction type, reported by Professor Stodola.[1] At the load showing the best economy, 2058 kw., the steam consumption was 15.21 lb. per kw-hr.; speed was 1501 r.p.m.; absolute steam pressure, 178.1 lb. per sq. in.; superheat 182 deg. fahr.; vacuum referred to 29.92 in. barometer was 28.41 in. (0.740 lb. per sq. in. absolute pressure). Calculated from these data the b.t.u. per kw-hr. on the basis explained in Par. 69, are 18,800; the heat utilized per pound of steam 224.5 b.t.u.; the available heat per lb. of steam 380.1 b.t.u.; and the efficiency ratio or the efficiency of the Rankine cycle is 59.1 per cent. The generator efficiency was low for a machine of this size. This is certainly a reliable test of a presumably well-designed turbine of the Curtis-Parsons type, and the results are not particularly favorable when compared with the data ordinarily published by manufacturers or their agents. It should be clear, therefore, that the adoption of a design consisting both of impulse and reaction stages along with excellence of constructive details does not necessarily give higher efficiencies of the Rankine cycle than other types. The Sulzer turbine, however, was laboring with one disadvantage in that it had three rows of moving blades and two rows of stationary blades instead of the modern Curtis construction employing two rows of moving and one row of stationary blades per pressure stage. Furthermore, it has

[1] Zeits. des Vereines deutscher Ingenieure; October 28, 1911, p. 1799.

never been shown that anything is gained by attaching to a blade wheel more than two rows of moving blades or buckets.

The writer felt that tests reported as showing Rankine efficiencies greater than about 68 per cent should be accepted only with a great deal of caution. To be more definite, he was not sure that the data of the first, third and fifth tests reported in Table 2 might be accepted without reservation. All of these tests are nothing more than manufacturers' statements, not properly verified. At a glance one might suppose that the second of these tests, taken from a descriptive article in the Zeitschrift des Vereines deutscher Ingenieure, came with some basis for consideration, but the test itself does not bear the earmarks of one made with very much accuracy. Practically all the data are given in round numbers, usually indicating approximation. As stated, the load is exactly 6000 kw., the steam pressure is given roughly at 13 atmospheres absolute, the steam temperature averaged apparently exactly 300 deg. cent. at the throttle throughout the test, and the steam consumption per kw-hr. was calculated not much closer than the nearest half-pound. As regards the fourth test cited, the writer is inclined to agree with Mr. Emmet in his statement at a previous meeting of the Society when this test was being discussed, that the "data are not altogether convincing." It seems to me that we should be very careful about incorporating into the Transactions reports of tests of boilers, engines and turbines supported only by the claims of manufacturers. It is very desirable that such tests should be made and reported by well-known engineers who have no financial interest in the business.

The statements in Par. 10 are not very clear as regards erosion or cutting of the blades. There is very little erosion of the blades in either the Curtis or the so-called Curtis-Parsons types if constructed of what are now considered by experts as suitable materials. The constant erosion of *continuous operation* for five years will have very little effect on the economy of the turbine as compared with the corresponding deterioration of a reciprocating engine.

The writer objected to another manufacturer's claim as given in Par. 51, where it is stated Mr. Zoelly "claims he gets the best results on his turbine by throttle governing." The Zoelly method of governing is cheap and simple, and that as the writer understands it, is the reason for its adoption. The writer would like to be shown, however, by authentic tests that it is even nearly as good as some other types.

C. V. KERR. In Par. 10 there is a statement as to the wear of the blading in the lower stages being due to the wetness of the steam. In the double-cup form of turbine in which the writer has been interested, the wear of the blading was found to be in the upper stages. This may be due to the fact that a greater proportion of the available energy was allowed to produce velocity in the upper stages, with the idea that part of it would be transferred to the lower stages, and thus equalize the power. Or it may be that the erosive power of the steam is greater in these upper stages, due to its greater density. It is a question whether it is possible to equalize this wear throughout the stages by an exact division of the available energy.

In his connection with the small turbine business the speaker had to do with the driving of auxiliary generators, pumps and blowers. He took the stand that his company wanted to know the maximum power to be developed by a turbine, then they would on their test put into the turbine a little more than that maximum power to make safe. It is very important for that class of work and he thought it should be for generator work also. A generator is supposed to stand a certain normal overload for a certain period. What is the use of putting into a turbine more power than is required to move the generator at that power and thus cause a burning out or other difficulties? The small steam turbine builders dropped into the habit of selling their turbines on the basis of power developed. With a casing of a given size, however, the power may be doubled or quadrupled by adding nozzles. But the small steam turbine is in competition with the steam engine and the electric motor, and they are generally sold on the power basis. It is the practice now on the part of at least two of the small steam turbine builders, to establish a minimum price for the turbine based on manufacturing cost and a rating upward on the basis of additional power developed.

Regarding the manufacturers' tests, there are members of the Society engaged in the manufacturing business whose study of testing methods is as intense and of as high an order as any who are doing testing. Questions are being raised as to standard methods of testing which come from manufacturers' engineers engaged in testing. There is no monopoly of engineering honor in one particular section of engineers. Although a manufacturer's engineer, the writer stands in favor of accepting all engineers' tests on their full value, for it is a mark of engineering

honor and a recognition of honor in others. He also believes that it is a mistake on the part of any engineer to put out data that are tainted in any way, since this will react on the offender sooner or later.

GEORGE A. ORROK referred to Professor Moyer's expressed doubt of the figures given in Table 2 of the paper for the Erste Brunner turbine. The results had seemed to him to be extremely good, and he had investigated them by writing to the engineer who made the tests, Professor K. Koerner, rector of the German Technical High School at Prague, who vouched for the data.

CLARENCE P. CRISSEY.[1] The accepted practice is to divide turbines into two fundamental types, namely, reaction turbines and impulse turbines. While these names may not be free from criticism, they are well understood and seem preferable to the names Parsons, Curtis and Rateau. While the latter terms give credit for inventions or variations in design, they are not based upon fundamental principles and, therefore, seem undesirable.

If it is desired to subdivide impulse turbines according to the manner of utilizing the steam, the following classes are suggested:

a Impulse turbines with only one velocity stage in each pressure stage

b Impulse turbines with two or more velocity stages in each pressure stage

c Impulse turbines in which all pressure stages do not have an equal number of velocity stages

Classification of impulse turbines might be carried further in order to illustrate a point of construction often overlooked: that is, the method of conducting the steam leaving the last buckets to the succeeding nozzles. Various constructions are:

d Impulse turbines in which the steam leaving the last moving buckets passes directly from the bucket into the succeeding nozzles without great loss of velocity

e Impulse turbines in which the steam leaving the last moving buckets passes into the casing and after losing practically all of its velocity enters the next set of nozzles

f Impulse turbines in which the steam leaving the last

[1] General Motors Co., Detroit, Mich.

moving buckets is utilized in some pressure stages according to the method described in *c* and in other pressure stages according to the method outlined in *d*

It follows from this classification that Rateau turbines fall in class *a-d;* Zoelly turbines fall in class *a-d;* Curtis turbines of the General Electric Company fall in class *b-d;* Curtis turbines of A.E.G. (large size) fall in class *c-f*.

The Zoelly turbine is similar in principle to the Rateau and it seems preferable to class the Zoelly turbine as a Rateau type just as the Allis-Chalmers is called Parsons and not Fullager.

It will be noted that in the paper the single-row stages of the A.E.G. turbines are said to be of the Rateau construction. The Allgemeine Electricitäts-Gesellschaft, however, state that these stages are constructed in accordance with the 1896 patent description of Curtis which represents the immediate forerunner of the Rateau and Zoelly constructions. The difference between the A.E.G. single-row stages and those of Rateau consists in the manner in which the steam leaving the moving buckets is utilized. This difference has been noted in the classification of turbine types given above, that is, the A.E.G. single bucket row stages are of type *d*, in which the steam leaving the moving buckets passes directly into the succeeding nozzles without great loss in velocity while the Rateau fall under type *c*, in which the steam is brought to rest before entering the next set of nozzles. The large A.E.G. turbines, therefore, appear to be of the Curtis type throughout, although not altogether of the type popularly known as Curtis.

The moving buckets of reaction turbines have in practice been supported upon drums and those of impulse turbines usually upon disks. The manner of construction does not, however, change the fundamental type and, therefore, the author's designation in Par. 24, of the Belliss & Morcom turbine as a Curtis-Parsons construction is confusing because this turbine operates entirely upon the impulse principle.

In reading Par. 5*b*, the impression might be gained that high steam velocities are inherent with impulse turbines having two or more velocity stages. Such is not necessarily the case. If low bucket velocities are employed the spouting velocity of the steam is also low. Impulse marine turbines illustrate this point.

It might be inferred from the last part of Par. 12 that constructions such as the A.E.G. sacrificed efficiency to first cost.

According to Table 2 the A.E.G. design is more efficient than the Rateau.

In Par. 22 reference is made to Fig. 3 showing a turbine with six pressure stages. It is stated that the A.E.G. build similar turbines up to 1000 kw. capacity. While the A.E.G. machines below 1000 kw. have two velocity stages in a pressure stage the writer understands that they have only two pressure stages.

It would be interesting if Professor Christie could tell whether Mr. Zoelly advances any reason for his claim of greater efficiency with throttle governing.

In connection with Par. 35 it might be observed that theoretically correct action of the steam in any impulse turbine occurs only at one set of conditions. For any variation in conditions or quantity of steam flowing a variation from the theoretically correct action must occur. The flat economy curves of impulse turbines prove that this is not important from the standpoint of efficiency, and practice shows that the end thrust is not appreciably increased by departure from the theoretical conditions.

The drum construction in impulse turbines was probably first used by the A.E.G. in the turbines for the steamship Kaiser, which was tested in 1905. This firm's experiments led them to state that the drum construction was advantageous, for the low bucket speeds existing in direct-connected marine turbines when the steam had expanded to 50 times its initial volume.

Using the fundamental classification given in this discussion there is only one type of "modified" or "combined" turbine, that is, the impulse-reaction machine.

It is interesting to examine the evidence as to which of the fundamental types is at present gaining in popularity. There is conclusive evidence that a large number of manufacturers formerly producing reaction turbines have adopted impulse blading in the high-pressure sections of their machines but there is little to indicate that the manufacturers of impulse turbines generally contemplate introducing reaction blading.

Par. 105 states that "several impulse turbines have recently been built in Europe where the expansion was not complete in the nozzles, so that a portion of the expansion took place in the first moving blades." The only turbines of this character which the writer knows of were abandoned and if others are being built they probably are more or less of an experiment.

In Par. 70 the following statement is made: "It is therefore

apparent that the efficiency ratio alone will express in the best manner the degree to which the designer has approached ideal results in his turbine." While the efficiency ratio comes nearer than any other single figure to expressing the economic results it must be used with caution because:

a The steam conditions must be taken into account. The same steam turbine operating with an equal amount of available energy will not have the same efficiency ratio if the energy is obtained in one case by a moderate vacuum and high superheat, and in another case by a high vacuum and low superheat.

A non-condensing turbine may have the same or a better efficiency ratio than a condensing machine, but this is no indication that the non-condensing turbine is as suitable or better for a given plant than the condensing unit.

b It is more difficult to obtain a high efficiency ratio with a large amount of available energy than with a smaller amount. Referring to Table 2 it is seen that the 6000 kw. machine, built by Erste Brünner M.F.G., has an efficiency ratio of 71.3 and the heat available per pound of steam is 380.7 b.t.u. The A.E.G. turbine of about the same rating gives an efficiency ratio of 68.7, but the heat units available per pound of steam are 434.2 b.t.u. or about 14 per cent greater. If the latter machine were operated with steam having 380.7 b.t.u. per lb., there is little doubt but that the efficiency ratio would equal or approach closely that of the Erste Brünner turbine.

c The efficiency of the electrical generators is included in the efficiency ratios. This has considerable importance if the ratios are used to compare turbine types. It is stated in Par. 59 that European builders generally guarantee better generator efficiencies than American builders.

d The results may be affected by the inclusion or exclusion of the power necessary for field excitation and auxiliaries, as well as other variations in testing practice.

From a technical viewpoint the revolutions per minute of the turbines cannot be neglected in making comparisons upon which to base opinions as to types. Either type is benefited by an in-

crease of rotative speed and it is almost certain that in the future the speed of all types will not differ materially for machines of a given rating and service.

Too much reliance must not be placed upon efficiency ratios, either in the selection of a turbine at the present day or as a basis of judgment regarding ultimate types because first cost, reliability, sustained economy, life and operating expenses are such important factors. The author points out, and it is well to emphasize the fact, that his figures for efficiency ratio do not represent operating conditions such as occur with varying loads or the average economy of any type of turbine.

E. D. DREYFUS emphasized the necessity of establishing some method of ratings which would be recognized by engineers at large, and also rules for conducting tests of this type of power generating unit, that is, including the turbine and generator as a whole. As the paper states the case there is now an undesirable dissimilarity in practice in rating this class of machinery.

The Society should lay down some guide or formulate some code which would create more uniformity and a better understanding in this connection. Failure to do this will leave room for criticism, which seems already to have shaped itself in the minds of some. It is well to note how this situation has been viewed by others, and the writer quotes a very brief abstract from an editorial from the Electrical Review and Western Electrician [1] under the heading of Ratings, which followed a paper he read one year ago on Turbines in which this vital question was seriously referred to:

> There are a variety of bases upon which ratings may be made. The ultimate capacity of the machine is one basis, the load at which maximum efficiency is secured is another, the load at which deterioration is less is another, the load which is fixed by some limitation of operation is another, and so on. Most electrical apparatus is limited in capacity by the allowable temperature rise and the rating is made upon this basis. The Standardization Rules of the American Institute of Electrical Engineers fix the allowable temperature, and performance specifications are drawn to coincide with these rules, or else specify limiting temperatures considered more suitable for the given working conditions. * * *
>
> Turning to steam engines we find the rating to be based upon operation at maximum efficiency and normal speed. * * * Practice is not entirely uniform here, however, and there is a certain leeway dependent upon

[1] October 14, 1911.

personal opinion or the characteristics of the particular engine. In any case, there is ample margin to take care of overload and a fairly definite general understanding as to what the rating signifies. * * *

With the steam turbine, on the other hand, there is no such general understanding and uniformity of practice, and it is not safe for the operating steam engineer to assume that a turbine with a given rating is capable of carrying the same overload that may be saddled upon the reciprocating engine of the same rating. * * *

* * * A uniform method of rating should be adopted just as quick as possible and nobody is in a better position to determine what that method should be than The American Society of Mechanical Engineers. By adopting such rules concerning steam machinery as the Institute of Electrical Engineers has done for electrical machinery, it will perform a real service to the country. * * *

CARL GEO. DE LAVAL. A new steam turbine has lately been designed on quite radical lines by Messrs. Ljungstrom, Stockholm, Sweden. It is a radial flow reaction turbine with steam entering between two disks passing from center to circumference between concentric blading rings. These two disks revolve with their shafts at the same rate of speed, but in opposite directions. At the end of each shaft an electric generator is attached. By doing this the relative speed of each blade becomes double the amount of the ordinary reaction turbine of the same number of revolution and diameter, with the consequent result that the blade rows become only one quarter as great. The general result is a turbine of small dimensions for the power and consequent smaller floor space even with the generators included than the present designs now on the market. Several turbines have been built and are being built.

The first turbine was 500 kw. at 3000 revolutions, which gave an efficiency value of 71.8 per cent and steam consumption 8.75 lb. per b.h.p. per hr., with steam of 175 lb. absolute and 250 deg. fahr. superheat and vacuum 28.5 in.

Sizes from 1000 to 7500 kw. with speed of 3000 revolutions have been designed on this double-rotating principle. Single rotating turbines of the same design have also been brought out, but with slightly less efficiency and increased steam consumption, the latter being about 10 per cent comparing the recorded efficiency ratio given by the author. Erste Brünner M.F.G. on a Curtis-Parsons is the highest recorded at 71.8 per cent, steam 156.2 lb., 1500 revolutions, lb. of steam per kw-hr., 13.82. The Ljungstrom 1000-kw. steam turbine gave 74.7 per cent efficiency ratio, taking steam in front of the throttle valve and 76.9 per

cent behind the throttle valve, steam 162 lb., speed 3000 revolutions, lb. of steam per kw-hr., 11.55 or 8 per b.h.p. It is stated that the critical speed of this turbine is 6300 revolutions, preventing the necessity as in many turbines of passing through the critical speed.

There are many novel details carried out in this turbine. The bottom is so shaped that it is placed directly on the condenser without the use of expansion joints. The two generators are supported on springs or slides on top of the extending ends of the condenser. The blades are of the Parsons type, milled from a solid bar of nickel steel, and having notched ends with blade roots turned over into the rings. In starting, the two sides of the turbine may not run at the same speed, but the exciting currents are strong enough so that at a certain speed the generators will synchronize automatically and run as one unit.

W. L. R. Emmet. In Table 2, which gives a comparison of various turbine performances, injustice is done to the Curtis type, and other incorrect impressions are given. This table is derived from various chance publications, but nothing is known concerning the authenticity of any of them. Such classification by merit as Professor Christie has made therefore ought not to be published without qualification.

The first machine mentioned in this table is a 2000-kw. Curtis-Parsons operating at 1500 r.p.m. and is reported to give an efficiency of 71.8 per cent with a moderate degree of superheat. Everybody familiar with the design of either Curtis or Parsons machines knows perfectly well that no such result has ever been produced under such conditions.

The efficiencies assigned to the five General Electric Curtis machines mentioned range from 63.6 per cent down to 61 per cent. The first, which operates at 3464 kw., is a representative result of the machine in question and is a good performance when the very high vacuum used is considered. Of the other four, only one of the tests is representative when the machine is correctly tested and in good condition, and in that case, also, the extremely high vacuum is the cause of a relatively low efficiency.

In discussing this table, the author states in Par. 70: "It is therefore apparent that the efficiency ratio alone will express in the best manner the degree to which the designer has approached ideal results in his turbine." To this statement exception should

be taken for the reason that it is naturally possible to produce higher efficiencies in turbine construction with moderate degrees of vacuum than it is with very high degrees of vacuum, the limitations to the effective use of high vacuum being many and their extent being affected by speed and capacity. Therefore the designer who produces a relatively low efficiency with a very high vacuum may have accomplished his purpose much more credit-

TABLE 3 TESTS OF REPRESENTATIVE CURTIS TURBINES

Customer	Date of Test	Load, Kw.	R.p.m.	Steam Pressure Absolute	Superheat, Deg. Fahr.	Vacuum	Water Rate, Lb. per Kw-Hr.	Efficiency, Per Cent.
		7526	720	184.40	134	28.65	13.732	66.6
Boston Edison Co.	1907	7481	720	185.10	129	28.58	13.866	66.6
Commonwealth Edison Co., Chicago	1907	12000	750	200.00	125	28.00	14.220	66.2
Boston Edison Co.	1912	12460	720	207.50	191	27.52	13.630	68.2
Stock	1911	4000	1800	189.00	0	28.00	16.050	64.3
Louisville Ltg. Co.	1911	6500	1800	180.00	100	28.00	14.780	66.10
British Thomson-Houston, Rugby (maker)	1912	2000	3000	165.00	150	28.00	14.400	67.3

ably than another who has produced a higher efficiency with a low vacuum.

None of the General Electric machines referred to in Professor Christie's tabulation, except the first mentioned, was designed later than 1904, while many of the other makes are of very recent production. Table 3 gives tests of a few representative Curtis machines. The first two tests apply to machines in Boston, very carefully and repeatedly tested when they were comparatively new. These machines are of the same date and type as most of those to which the figures in Professor Christie's tabulation purport to apply, and are representative of the original large 5-stage vertical units when correctly tested and in good condition. These tests were made with very perfect facilities by the Boston Edison Company's engineers.

The third test applies to an 8000-kw. machine in Chicago of the same date and type. One of the tests in Professor Christie's paper also purports to refer to this machine under different conditions.

The next three items refer to recent six-stage Curtis machines, and the last refers to a three-stage Curtis machine recently produced by the British Thomson-Houston Company. This latter machine produces a very remarkable result when its small capacity and extreme simplicity are taken into consideration.

The accurate testing of steam units is a matter which requires great care and is susceptible of many kinds of error. Many figures concerning such tests which are entirely incorrect are being constantly circulated. It is only by comparison of many tests and careful observance of consistency as to results and characteristics that one can ever be sure as to turbine economies. The figures given in Table 3 have been analyzed and compared and are unquestionably correct. It will be observed that they give an impression concerning the value of Curtis turbines which is very different from that produced by Professor Christie's tabulation.

The author's closure is to be published later—EDITOR.

A NEW ANALYSIS OF THE CYLINDER PERFORMANCE OF RECIPROCATING ENGINES

By J. Paul Clayton, Published in the Journal for April

ABSTRACT OF PAPER

Our knowledge of the cylinder performance of reciprocating engines is obtained almost entirely from indicator diagrams. These diagrams provide a measure of the work performed, thus enabling the efficiency or economy of the engine to be determined; they also provide an aid for setting valves, and furnish the basis in steam cylinders for applying Hirn's analysis, for measuring the initial condensation, and for finding the diagram factor for the purposes of design. It has been generally thought by engineers that the evidence contained in the diagram was limited only to these and minor uses.

The investigation described in this paper has disclosed the fact that the indicator diagram contains in itself the evidence necessary for an almost complete analysis of cylinder performance, the results of which have not heretofore been considered possible.

In obtaining these results the diagram has been transferred to logarithmic cross-section paper and thus a figure has been drawn which will be called a logarithmic diagram. By the aid of this diagram it has been found that the expansion and compression curves of all elastic media used in practise obey substantially the polytropic law $PV^n=C$. From this fact there have been developed rational methods of approximating the clearance of a cylinder, of closely locating the cyclic events, and of detecting moderate leakage with the engine in normal operation.

It has been discovered that the value of n in the law $PV^n=C$ is controlled directly in steam cylinders by the value of the quality of the steam mixture at cut-off, called X_c, and that the relation of X_c and n is practically independent of cylinder size and of engine speed for the same class of engine. This fact enables us to determine the actual amount of steam and water present in a cylinder at cut-off from the experimentally determined relations of X_c and n, and thus to obtain the actual steam consumed from the diagram.

The paper contains an exposition of the graphical methods employed, charts giving the relations of X_c and n for determining the steam consumption from the diagram, and examples of typical logarithmic diagrams.

The American Society of Mechanical Engineers, 29 West 39th Street, New York. All discussion is subject to revision.

DISCUSSION

ARTHUR L. RICE said that Mr. Clayton's paper for the first time gives the engineer a chance closely to keep track of what the engine is doing at all times while it is in actual operation. He had had occasion to use this method since the publication of the paper in The Journal. In the use of a high pressure of steam in the engine and a high speed, where the indicator card is a small one, it is absolutely essential that very great accuracy should be observed; otherwise the indications as to the amount of steam consumption may be quite misleading. As to the percentage of accuracy of this method as compared with weighing the steam from the surface condenser, Mr. Rice found from tests that the variation was from 2 to 6 per cent. The logarithmic curve may be plotted directly in inches, without reducing the volume to cubic feet. The only volume of interest is that at cut-off, which is a single volume and easily computed. It is, however, necessary in all cases to take care that the logarithmic paper used is accurate, which is often not the case with the paper in the market.

R. C. STEVENS disagreed with Mr. Clayton's treatment of the problem of leakage, and questioned where the author takes into account the leakage of steam directly out of the exhaust, or that part of the steam which enters the steam chest but, by reason of the valve leakage, passes directly out of the exhaust without entering the cylinder. That this loss is enormous, even in many new engines, is too little appreciated. For example, in a case quoted by Mr. Stevens from an article by George Mitchell in Power, October 11, 1910, it was as high as 22 per cent in an engine specially fitted by its makers with a new valve. What is true of piston valves is doubly true of the flat balanced valve with pressure plate. Test on one such engine of 80 h.p. capacity which the owners considered in good condition showed 624 lb. steam leakage per hour, or about 21 h.p. wasted, or over 25 per cent. leakage. It is common practice to allow 0.003 to 0.004 in. freedom between the valve and pressure plate when newly fitted, and in operation the valve wears small and the seat wears large. Altogether, valve leakage is far more important than piston leakage, which may be almost negligible for years. Mr. Stevens therefore challenged Mr. Clayton's assertion that steam con-

sumption may be approximated from indicator diagrams to within an average difference of less than 4 per cent from the test results, and only in rare cases as much as 8 per cent. For, if 20 per cent or more of the steam that enters the throttle goes out of the exhaust without making its presence felt in any way on the indicator, it is difficult to understand how any analysis of the indicator diagram or its derived logarithmic diagram can establish the actual steam consumption of an engine to within even 20 per cent of accuracy.

SANFORD A. MOSS pronounced the paper to be the first new thing on steam engines for many years, but insisted on the necessity of taking leakage into account. Steam engines are seriously hampered by leakage and when this occurs Mr. Clayton's method cannot be entirely depended upon, unless suitably modified. In the paper an attempt is made to give a variation of the law with pressure, and it did not seem to him that the tests warrant any distinction between different pressures and that the author should use the equation from the average of all pressures rather than the chart of Fig. 6.

R. C. H. HECK. In the logarithmic diagram as a means of investigation, in the proposition that exponent n is practically constant for the respective curves of a particular steam diagram, and in establishing a definite relation between n and quality at cut-off, this paper makes three distinct contributions to the field of research and knowledge which lies between the performance of the actual engine and that of the ideal engine with a non-conducting cylinder. In this region, rational deduction from simple premises is not available, chiefly because it is not possible to simplify and make definite the needed premises: consequently, theory can be built up only by such formulation of empirical results as is here undertaken.

To say that n varies widely with different conditions of working is merely to give special form to a statement of long recognized fact. We know that the approximation of actual expansion and compression curves to the equation $PV = C$ is very rough. A common range of variation with saturated steam is from 5 to 10 per cent, while in extreme cases the product PV departs a good deal farther from constancy. Nevertheless, the hyperbola has been and will continue to be a useful reference

curve, not from any sanction in theory, but simply because it represents constant PV.

Critical judgment upon the method derived and proposed for estimating actual steam consumption could be given only after extensive trial and use. The showing in Table 4 seems to be very favorable. Manifestly, the scheme is as yet incomplete, since the service chart, Fig. 6, is limited in application to one exhaust pressure. In working along the older line of a formula containing size, speed, cut-off ratio, and range of steam tempera-

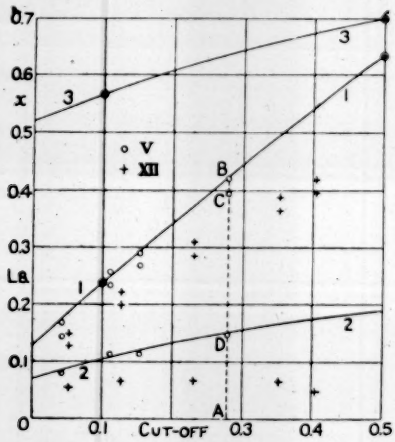

FIG. 26 PLOT OF STEAM QUANTITIES PER REVOLUTION ON CUT-OFF AS BASE, FROM TEST SERIES V AND XII, TABLES 1 AND 2

CURVE 1, ORDINATE *AB*, TOTAL STEAM PRESENT DURING EXPANSION, MADE UP OF MEASURED CONSUMPTION *AC* PLUS CLEARANCE STEAM *CB*.

CURVE 2, ORDINATE *AD*, MISSING STEAM AT CUT-OFF, FOUND BY SUBTRACTING INDICATED STEAM *BD* FROM TOTAL STEAM *AB*.

CURVE 3, CUT-OFF QUALITY x_c, CORRESPONDING WITH CURVES 1 AND 2.

CURVES ARE DRAWN FOR THE SATURATED STEAM TESTS ONLY; EXTENSIONS TOWARD THE RIGHT ARE GUIDED BY THE FORM OF OTHER SERIES, WHICH HAVE LONGER RANGES OF CUT-OFF; THE CIRCLED DOTS ON ORDINATES AT 0.1 AND 0.5 CUT-OFF SHOW VALUES USED IN THE EXAMPLES; THE SUPERHEAT TESTS ARE HERE GIVEN FOR PURPOSES OF COMPARISON ONLY.

ture as elements, the writer has consistently found the last the most difficult to represent satisfactorily.

It is a matter of considerable interest to know just what sort of thermal interactions are involved in the exact production of the curve $PV^n=C$ with a steam and water mixture. General relations cannot be expressed in serviceable shape, therefore it is necessary to resort to the solution of numerical examples. Taking typical conditions determined by test series V, Table 1, as

plotted in Fig. 26, the writer has calculated results which are presented graphically in Figs. 28 and 29.

The controlling conditions for this pair of examples are as follows:

General:

Pressure at cut-off, lb. absolute	111
Corresponding steam temperature, deg. fahr	335.5
Piston displacement per revolution, cu. ft	3.10
Clearance, per cent	7.5

Particular:	Case *A*	Case *B*
Quality at cut-off, x_c	0.5667	0.7000
Exponent n	0.92	1.025
Weight of steam mixture, lb	0.2375	0.6332

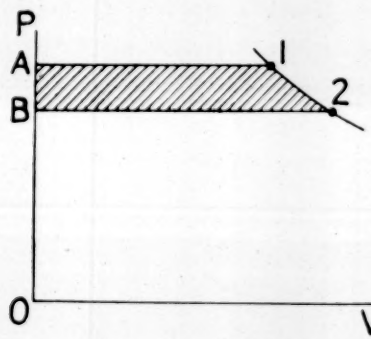

FIG. 27 DIAGRAM SHOWING METHOD OF CALCULATING RESULTS IN FIGS. 28 AND 29

Thermal quantities were first worked out for the weights of steam just named, then were reduced to b.t.u. per cu. ft. of piston displacement for use in the diagrams.

The method of calculation is illustrated by Fig. 27 and the relations involved are expressed by the equation

$$h_1 + q = A \int_1^2 v dp + h_2 \quad \ldots\ldots\ldots\ldots [4]$$

Here h_1 and h_2 are heat contents or total heats at the beginning and end respectively of the expansion interval, or at points 1 and 2; q is the heat imparted to steam by cylinder walls; and

$$A \int_1^2 v dp = \frac{144n}{778(n-1)} (p_1 v_1 - p_2 v_2) \quad \ldots\ldots\ldots [5]$$

is the heat equivalent of the work done in expansion from p_1 to p_2, represented by area $A12B$. The scheme is to use temperature intervals, of 15 deg., passing from t to p through the steam table.

In Fig. 28, distances from the base to line 1 represent heat used in the performance of external work, calculated by equation [5]. Distances from 1 to 2 represent changes $(h_2 - h_1)$ in heat content, positive in case A where h_2 is greater than h_1 $(n =$

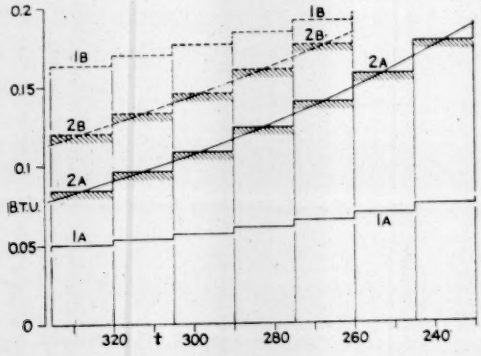

FIG. 28 RATES OF HEAT RECEPTION PER DEGREE OF CHANGE OF STEAM TEMPERATURE

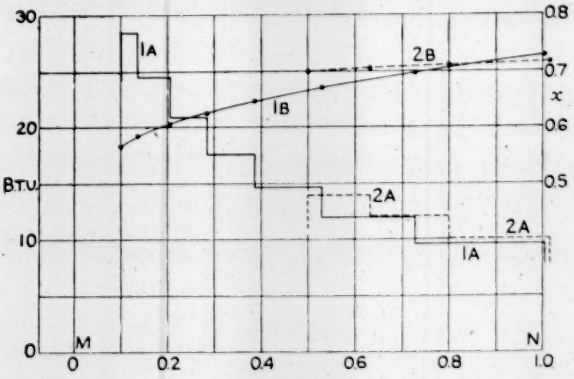

FIG. 29 RATES OF HEAT RECEPTION PER CUBIC FOOT OF PISTON DISPLACEMENT, ALSO CURVES OF STEAM QUALITY

0.92), negative in case B where h decreases with expansion $(n =$ 1.025). The net or resultant distance from base to line 2 shows heat received q.

Strictly, the areas beneath the several sections of the broken lines in Fig. 28 represent the heat quantities just named, while to the scale of the diagram the ordinates show rates of heat conversion, change, or transfer per degree of change of steam tem-

perature. Thus in case A, with 0.2375 lb. of steam, the heat added from 305 to 290 deg. is 5.05 b.t.u. This is divided by 3.10 to make it correspond with the steam in action per cubic foot of piston displacement and by 15 to make it heat per degree. In case B, with 0.6332 lb. of steam, the similar heat quantity is 6.76 b.t.u., to be reduced to the terms of the diagram by the same division. Because thus related to steam quantities other than 1 lb., the ordinates of curves 2A and 2B are not identical with specific heat; but they are proportional to unit heat capacity.

Of course, for this operation of hyperadiabatic expansion, the specific heats represented by curves 2 of Fig. 28 are negative. But the point of special interest here brought out is their increase with fall of temperature.

In Fig. 29 base MN represents 1 cu. ft. of piston displacement, with clearance OM added at the left; and it may also be taken to represent stroke of piston. The ordinate added is now, heat added divided by change of volume in expansion. Thus for the interval from 305 to 290 deg. in case A, heat q or 5.05 b.t.u. is divided by a volume change of 0.2417 cu. ft., and the quotient 20.9 is an ordinate of line 1A.

As bearing on the maintenance or production of a constant exponent n, we note that for case A, with early cut-off, a very rapid heat reception is required at first; but with later cut-off the requirement is more nearly uniform. Now the store of latent heat in the steam is relatively so large that a very small change in the manner of variation of quality x will set free or absorb quantities of heat which are of the same order of magnitude as those calculated for Fig. 28. We may therefore conclude that there is nothing in the showing of Fig. 29 to antagonize the empirical fact that for ordinary conditions of working n is practically constant. But we can see that for very early cut-off heat is likely to be less rapidly transferred at first than is required to produce the average value of n, while later the requirement may be exceeded. Such an action, with n at first large and pv decreasing, followed by decrease of n and increase of pv, is just what is observed in this case.

A few classic examples from our records are set forth in Fig. 30. The first two curves show the smooth form of the polytropic in the scheme of coördinates which is used. No. 3 is a case of very low speed, with rapid growth of pv at low pressure and accompanying decrease of n. No. 4 shows the opposite ex-

treme of high superheat, with fair approximation to a value of n considerably greater than unity. In the engine of No. 5 release was late, so that the expansion curve ran clear to the end of the stroke; and in this period of very slow piston movement there is time for a large inflow of heat, strengthened by jacket action. Curve 5 shows, then, condensation continuing past cut-off and very marked reëvaporation at the end of expansion. It

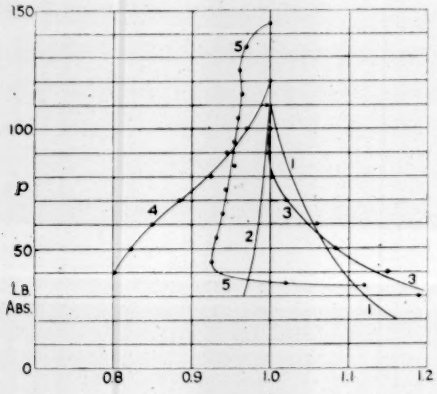

FIG. 30 PLOTS OF pv ON p, REFERRED TO THE INITIAL VALUE OF pv AT CUT-OFF AS UNITY

CURVES 1 AND 2, FROM EXAMPLES WORKED AND ILLUSTRATED IN FIGS. 28 AND 29 VALUES OF n=0.92 AND 1.025, RESPECTIVELY.

CURVE 3, TRANSACTIONS, VOL. 10, PAGE 742, FIG. 170, CYLINDER 17 IN. BY 30 IN., SATURATED STEAM, SPEED 8.64 R.P.M.

CURVE 4, TRANSACTIONS, VOL. 25, PAGE 267, CYLINDER 16 IN. BY 42 IN., SUPERHEAT 352 DEG. AT THROTTLE, 187 DEG. IN CYLINDER AT CUT-OFF, SPEED 103 R.P.M.

CURVE 5, TRANSACTIONS, VOL. 16, PAGE 176, LEAVITT PUMPING ENGINE AT LOUISVILLE, CYLINDER 27 IN. BY 120 IN., SATURATED STEAM, FULL JACKETS, SPEED 18.6 R.P.M.

is a striking example of how decidedly steam may fail to conform to the polytropic equation.

F. W. MARQUIS discussed the method of Mr. Clayton as a valuable aid in connection with the study of locomotive performance. It has heretofore not been possible to determine the steam consumption of locomotive engines under certain predetermined conditions of speed, cut-off, etc., by means of road tests, since in road service these conditions cannot be maintained constant for a long enough period to constitute a test. Mr. Clayton's method gives a way to determine the steam consumption under any conditions of speed, cut-off, etc., at which a locomotive can be oper-

ated, by means of indicator cards which can be taken while the locomotive is being used in regular road service, and a study of the effects of speed, cut-off, steam distribution, etc., can now be made by means of data obtained on the road tests. Further, from the logarithmic indicator diagram it is possible to determine accurately the times at which the different events of the stroke take place. This furnishes a comparatively simple method for determining the spring in locomotive valve gears. A study of this spring and its effect upon steam distribution, the stresses in valve gear parts, steam consumption, etc., with different types of valve gears, and under different conditions of running, is sure to prove of great value. The same method may also help to determine exactly why it is harder to make schedule time with passenger trains in cold weather than in warm.

Train resistance is greater in cool than in warm weather; a locomotive cannot make as much steam in cold weather, and again, steam must be furnished for heating the train. How much steam does it take to heat a train in cold weather? The steam thus used does not pass through the nozzle and help produce draft to burn the coal, as the steam used by the cylinders does. Is the amount large enough seriously to affect the steaming qualities of the locomotive? Information upon the above questions appears to be very meager, and has not in the past been easy to obtain. Now, however, it is possible to determine from indicator diagrams how much steam is used by the cylinders while the total steam used may be determined by means of tank measurements. The difference will be the steam used in heating and lighting the train, in running the air pump, etc.

Mr. Marquis called attention to the fact that the present paper includes data from only one locomotive test and said he understood that other data were to be given later.

C. D. Young. Good use has been made of the fact which has been established for some time that a practically constant quantity is obtained in the product of the absolute pressure by the nth power of the volume of steam under compression or expansion, the volume including the clearance space. This law has been well known mathematically as a straight line when expressed by its logarithmic equation, but Mr. Clayton has succeeded in defining what n stands for, showing that it is a quantity definitely related to the quality of the steam, and for any

given engine working under given conditions the value of n depends on the quality of the steam.

The writer assumed that this followed Mr. Clayton's discovery that the expansion and compression curves of indicator diagrams do as a general rule develop straight lines when absolute volumes are plotted against absolute pressures on logarithmic paper: and that where the plotted lines are not straight some defect in the cylinder is indicated and the amount of the discrepancy becomes an indication to some extent of the magnitude of the defect.

The writer feels that Mr. Clayton's discovery will develop to a point where it will be exceedingly valuable as a short-cut method and he is to be congratulated upon opening up a new field for study of cylinder performance of reciprocating engines. There are a number of statements in the paper, however, which he is inclined to question. For example, in Par. 8 is the broad statement that since working with the new method of analyses existing methods have been found to be in a very crude state and many deductions from them are without foundation or meaning. He finds nothing in Mr. Clayton's paper to substantiate such a statement and would hesitate broadly to criticize the past work on engine performance and deductions which have been made in that light of results developed by Mr. Clayton.

Again in Par. 11 statements are made to the effect that facts as set forth were unknown previous to this method of investigation: where as a matter of fact the writer believes that all the statements have been confirmed by work of others on this same subject.

The statement hardly seems warranted in Par. 74*c*, in which Mr. Clayton says that this method is more accurate than the average test, and is the only accurate method available for testing certain classes of engines. Assuming that the cards have been taken correctly and represent the actual cylinder performance, and before it is definitely determined what the actual water rate of cylinder is, the steam, either entering or leaving the cylinder should be measured by such means as are now available.

From the data obtained in Mr. Clayton's tests it is quite evident that the quality of steam mixture at cut-off when using superheated steam is in a large number of cases below the condition of dry and saturated steam and in Table 2, column 13, the quality of steam is given as below unity. The writer hesi-

tates to criticize this figure, although he possesses data which would indicate that the quality at cut-off as given by him for superheated steam will not be as low as that which he has given and obtained graphically.

Until Mr. Clayton's method can be checked against some of the accurate tests which are daily being conducted at Pennsylvania Railroad Company's Locomotive Testing Plant at Altoona, where all conditions can be controlled and accurate measurements taken, the writer hesitates to suggest further errors in the data as presented.

J. B. Stanwood. This paper is a good example of what an advanced scientific treatment of a well-known subject may develop. While the logarithmic treatment of expansion curves is not new,[1] the author has discovered that a relation exists of a very simple character between the quality of steam at cut-off in a steam engine and the expansion curve which develops from this quality of steam. This is an interesting fact which may prove of considerable value in further analysis of the action of steam in the cylinders of reciprocating engines.

In regard to the actual deductions made Mr. Stanwood calls attention to the fact that Mr. Clayton's deductions are based almost entirely upon experiments made with a 12 by 24-in. Corliss engine, and believes that the applicability of his method to the determination of the actual values of initial quality of the steam can be determined only after comparing a number of experiments of the same character on different sizes of engines. In the instance at hand, leaving leakage out of account, it would seem that the initial quality of steam at cut-off is dependent upon two things: the amount of steam actually condensed up to cut-off, and the quality of steam in the clearance space just before the admission. If the quality is based simply on the difference between steam as accounted for by the indicator at cut-off, and the water consumption as determined by the amount condensed in the condenser, he fears the result is not correct, as no consideration has been taken of the quality of the "cushion" steam. It may be possible that the quality of cushion steam may be as low as 10 per cent, especially if Mr. Willan's

[1] See Prof. Perry in the discussion of Mr. Willans' paper on Non-Condensing Engine Trials; Proceedings of the Institute of Civil Engineers of Great Britain, vol. 43, 1888.

theory is true that water retained on the clearance surface forms an important factor in creating cylinder condensation. Professor Zenner has stated that if he knew absolutely the quality of steam at any point during the compression he could determine the steam consumption of a steam engine from its indicator card (leakage being neglected). This question of the quality of cushion steam may have something to do with the variations in the results obtained by Mr. Clayton in his comparison of different engines, as shown in Table 4.

There are probably few, if any, engines operating without leakage and the writer thinks there must have been quite an appreciable amount in the 12-in. by 30-in. engine tested by Mr. Clayton. For instance, in Table 1, with 109 lb. absolute pressure at the throttle and 25 per cent cut-off, this engine is stated to have a quality of steam of 58.8 per cent. This means a cylinder condensation of 41.2 per cent, which is high for this cut-off in comparison with most tests that have been made on Corliss engines. George H. Barrus in his book on Steam Engine Tests gives the result of his experience with these engines and he estimates as a result that the amount of condensation for 25 per cent cut-off is on an average about 24 per cent. If therefore there was no excessive leakage in this engine, then these experiments show a remarkable difference between condensation in small engines and condensation in larger engines of the same general type. In Table 4 there is recorded a test (Analysis No. 107) of a 16-in. by 42-in. Corliss engine, in which the quality of steam is 77.4 per cent. In the same table, with a Corliss cylinder 23 in. by 60 in., the quality of steam is 76.7 per cent. The writer suspects that these engines were among those tested by Mr. Barrus in his work entitled Steam Engine Tests, and if so, the cut-offs were about 25 to 30 per cent. Is there this difference between engines of 12 in. and 16 in. and 23 in. bore? If not, there must have been some other cause which would make the quality of steam so low in the 12-in. by 24-in. engine and the writer suspects that it is leakage.

The discovery of Mr. Clayton may lead to the solution of some problems heretofore unsolved, such as determination of the actual quality of steam in the compression spaces; prediction of steam consumption, especially as influenced by the size of cylinders; effect of clearance and clearance spaces on the economy of the steam engines. Designers may also be able to improve the

proportion and construction of steam cylinders and valve gears by the character of the expansion curve under different conditions analyzed by this logarithmic method. The writer fears that a determination of the steam consumption of many types of engines by this method would not be very reliable. There are so many cylinders in which the steam is short circuited directly from the steam side of the cylinder to the exhaust side of the cylinder that no evidence of this loss appears on the diagrams, so that the use of Mr. Clayton's method would not be at all satisfactory. There are many single valve engines with balanced valves in use in this country in which the leakage begins to increase from the day they go into service; any diagram taken from them without absolute knowledge of the valve conditions would be useless, and we would caution engineers as to the unreliability of this method for engines of this class. When valves are made tight and can be kept tight, the investigation along these lines may be helpful.

William D. Ennis was able in one or two instances to verify Mr. Clayton's law. A small automatic engine gave $n=0.96$, with a dryness at cut-off of 0.62. A Corliss engine gave $n=1.02$ average corresponding with an indicated steam rate of 26 lb. per i.h.p.-hr., noncondensing. A study of values of n for the compression curve is interesting. Values exceeding 1.0 have repeatedly been found, along with such absence of high final temperatures as demonstrates that the customary Hirn assumption of initial dryness equal to 1.0 must be incorrect. Professor Ennis has found in two cases, with small automatic engines, n for compression=1.85 and 1.80 and in another instance it was apparently around 4.0.

From the thermodynamic side, the paper is of particular importance from the new light it throws on the question of heat interchange. Textbooks state that during the early part of expansion the steam continues to give up heat to the cylinder walls; afterward, as its temperature falls, again receiving this heat. But if the expansion line of the steam diagram conforms to the polytropic $PV^n=C$ there can be no such reversed transfer: the steam must be either steadily gaining or steadily losing heat. And, according to Mr. Clayton, it loses or gains according as it is dry or wet at the point of cut-off. In the light of the present paper, Professor Ennis is inclined to think that

the apparent reversal of transfer has been generally due to leakage. In the Callendar and Nicholson experiments the steam temperature dropped to that of the metal of the cylinder head in 1/18 sec. after cut-off. Considering the lag and dampening of wall temperature fluctuations, the conditions thus favor a gain of heat by the steam practically throughout expansion. Strangely enough, also, the wall temperature taken at a point in the side wall just inside the face of the head, began to fall while the walls were about 17 deg. fahr. cooler than the steam.

The subject of steam leakage is one of the most important which the paper suggests. The diagram attached is so clearly like some of those embodied in the paper in its indication of leakage that it is worth including for its confirmatory value. Leakage between two chambers will, other factors being constant, be proportional to the pressure difference and inversely proportional to the seal of the valve. The diagram sketches these variables and shows that leakage is most likely to occur:

a through the *steam* valve, just after cut-off, or just before admission

b through the *exhaust* valve, during the early part of a stroke

c through the *piston*, during the early part of a stroke.

In cases (*b*) and (*c*) the diagrams are scarcely likely to show leakage until after cut-off has occurred. Leakage through the steam valve, occurring just after cut-off, may be shown by a steam chest diagram. It raises the expansion curve during the early part of expansion. When it ceases, the true polytropic curve begins. Pre-admission leakage at this point raises the compression curve, and may take place at such a rate as to give a curve apparently polytropic, with an extremely high n value. A steam chest diagram may confirm the diagnosis. Leakage through either piston or exhaust valve, occurring just after cut-off, *depresses* the expansion curve. If it is the piston that leaks, a corresponding *rise* of pressure may in some cases be noted on the exhaust line of the compression diagram for the other end of the cylinder. If such leakage occurs before cut-off, the diagram in question will not show it, but the companion diagram may show a rapid rise of pressure during the early part of its exhaust period. Mr. Clayton's paper is thus far but a beginning. The bearing of the new law on the application of Hirn's analysis and the temperature-entropy diagram to actual tests will com-

pel us to revise our judgment as to the utility of these forms of investigation. A new and undoubtedly fruitful field has been opened in steam engineering.

FORREST E. CARDULLO. It has long been known that the expansion of the working fluid in practically all reciprocating engines approximates the law $PV^n = C$, in which P is the absolute pressure, V is the absolute volume of the working fluid, and C is a constant. The paper before us proposes a most interesting method of cylinder analysis which takes advantage of this fact.

With that part of the paper which deals with the analysis of gas engines, air compressors and other reciprocating engines employing permanent gases, the writer is in complete accord. There are, however, many points in connection with that portion of the paper which deals with steam engines, with which he cannot agree.

One of the first questions which arises in this connection is how closely does the expression $PV^n = C$, represent the actual facts in the case. Taking the adiabatic expansion of steam and plotting the relation between the pressure and the volume on logarithmic paper, we find that the expansion curve is convex toward the origin, but that the curvature is so very slight as hardly to be noticeable. When plotted to the scale which Mr. Clayton uses and for the range in pressures employed in reciprocating engines, the adiabatic expansion line is sensibly straight.

In Par. 164, Mr. Clayton states that the only irregularities observed in the compression lines of steam cards occur with low speed, and are in the form of a "hook" which occurs at the point where the saturation temperature of the cushion steam rises above the temperature of the cylinder walls. The writer has seen such hooks due to cylinder condensation, in the compression curves of cards from a 7 by 12 straight line engine running at 300 r.p.m. These hooks will always occur with any type of engine when the compression is sufficiently great. They are more pronounced in engines having a small clearance volume, and the higher the rotational speed of the engine the higher the compression pressure at which they occur.

During the expansion the steam is condensed at the beginning of the expansion line unless the drop in pressure during admission is so great that the saturation temperature of the steam falls below the temperature of the walls before expansion commences. Later on, instead of condensation there is re-evaporation during the ex-

pansion period. If the adiabatic line is straight on the logarithmic plane, it follows that the actual expansion line cannot be, but will have a decided bend in it at the point where the saturation temperature of the steam is equal to that of the cylinder walls.

This effect is well shown in the temperature-entropy diagram of many steam engines which has the form shown in Fig. 31. In this figure, at the point marked *A*, the expansion curve is tangent to the adiabatic, and this is the point where the temperature of the steam becomes equal to the temperature of the cylinder walls. The temperature of the cylinder walls rises with the speed of the engine and consequently the bend in the logarithmic expansion curve will rise, until in the case of many high-speed engines it may be higher than

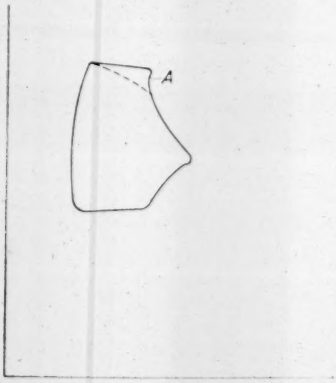

FIG. 31 TEMPERATURE-ENTROPY DIAGRAM SHOWING FORM OF EXPANSION LINE

the actual expansion line, which would be the case in Fig. 31 if the steam line was the dotted line instead of the full line.

The appearance of this bend is exactly the same as the appearance which Mr. Clayton attributes to leakage, and it may be questioned therefore whether leakage always exists when the expansion line has the form shown in Fig. 13-b.

In the case of a fluid expanding in a conducting cylinder, n is never a constant, but is a variable, although the variation is usually very slight and nearly uniform. Any appreciable irregularity in the logarithmic expansion or compression line is therefore evidence of leakage or of some other disturbing condition. We may therefore expect that a further study of the logarithmic expansion and compression lines will make clear some very interesting facts in connection with reciprocating engines.

Par. 42 states that the skin surface of the cylinder must be heated once every cycle from the temperature acquired by contact with the exhaust steam nearly up to the temperature of the admission steam. This statement is misleading since the range in temperature of the metallic surface is known to be comparatively small, being only from a fifth or a tenth of the range of the temperature of the steam.

From Par. 47 it is inferred that the phenomena occurring after cut-off are independent of the form of the cylinder and the relation between the volume and the clearance surface. The writer cannot believe that this is quite true. By changing the quality of steam, the volume, the clearance surface and the speed, there may be several engines having the same quality of steam at cut-off, but in which the rate of radiation of heat from cylinder walls into the steam will be quite different. Under the circumstances different values of n may be expected for the same quality of steam at cut-off. The mere fact that two or three engines of apparently dissimilar design give substantially similar values for n and X_c does not disprove the possibility of this variation, and until this point receives further attention, it cannot be taken for granted.

In Par. 64 it is stated that this method is the only one which measures the *rate* of steam consumption of an engine. A steam meter may be employed for the same purpose, and will probably be more accurate. In certain types of engines and with certain forms of indicators the error of the expansion line due to vibration will make the method very inaccurate. I do not believe that the method is sufficiently accurate to warrant its use in determining steam consumption, although it will probably be very useful in determining losses from leakage and in showing whether or not an engine is operating under the most economical conditions possible.

The method does not take account of leakage and if there is any the steam consumption obtained will be lower than the true steam consumption of some unknown amount. Since there are forms of leakage which will not show when the logarithmic analysis is employed, it follows that there will be cases when the steam consumption obtained by this method will be too low and yet there will be no evidence of the fact. On this account, while the steam consumption may be determined with reasonable accuracy in many cases, one can never be sure whether the method is accurate or not, and it must therefore always be looked upon with suspicion.

In connection with Par. 74, how does this method check with itself when the steam consumption of the high-pressure and low-

pressure cylinders of a compound engine are determined independently by its use? Does it give the same steam consumption for both cylinders as it should? In the same paragraph the writer would call attention to the fact that the method cannot be applied when the load on the engine is varying, since the temperature conditions of the cylinder walls are those acquired from a previous revolution, and the quality at cut-off and the form of the expansion line will be different from what it would be were the engine operating with the same cut-off under steady load.

In Par. 84 is given a graphical method of determining the clear-

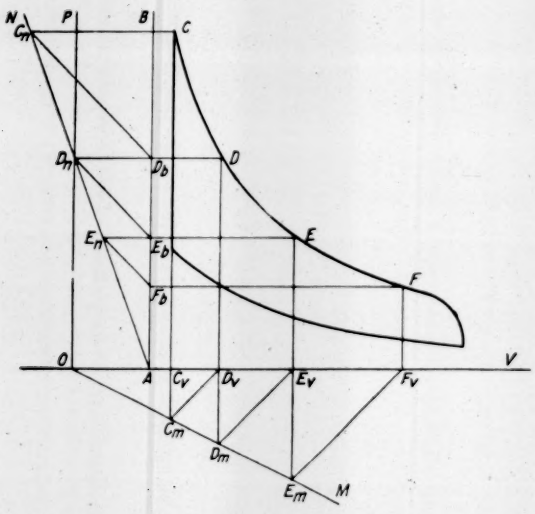

FIG. 32 GRAPHICAL METHOD FOR DETERMINING CLEARANCE

ance which is based on the assumption that the expansion line follows the law $PV^n = C$. The following is submitted as a preferable method:

In Fig. 32, CF is the expansion line of a gas engine card and OV the axis of zero absolute pressure. At any point A on this axis erect the perpendicular AB, and draw AN making any convenient angle to AB. Draw CC_n and draw C_nD_b making an angle of 45 deg. with AB. Through D_b draw D_nD and D_nE_b at 45 deg. Similarly, draw E_nE, E_nE_b, and F_bF. From F drop a perpendicular to OV and draw F_vE_n making an angle of 45 deg. with OV. From E_vD_m drop a perpendicular from D to determine D_m. Draw D_vC_m and drop a perpendicular from C to determine C_m. If the expansion line CF fulfils

the equation $PV^n = C$, $C_m D_m$ and E_m will lie on the same straight line and that line will intersect OV at the point of zero volume. Erect *OP* at that point which is the origin of pressure and volume. This method [1] is more elegant, more accurate, and more simple than the method proposed in the paper.

There is no question but what the logarithmic analysis will show evidence of leakage in many cases. In certain cases, however, when there are simultaneous leaks of inlet and exhaust valves, it is quite possible that no leakage can be detected. This is also the case where the leakage takes place by the bodily transfer of water under the surface of a slide valve, which is the form of leakage most common in high-speed engines. There is no possibility of approximating the amount of leakage from the logarithmic diagram. Data in regard to the relation of the quality of the steam at cut-off and the exponent of expansion, obtained from engines equipped with valves known to be subject to leakage, should be regarded with suspicion.

It is a well-known fact that an engine which is apparently quite tight when standing, may have considerable leakage when it is running. So far as I can find from the paper, no attempt was made to determine the running leakage. It is unfortunate that Mr. Clayton did not check the quality of steam in the cylinder at cut-off by the thermoelectric methods employed by Callander and Nicholson. He did not state whether the engines tested have running leakage or not. It is desirable to have more evidence on this point, especially before using the method, for the purpose of determining the steam consumption.

If experience justifies the claims for this form of analysis, indicators which will draw the logarithmic diagrams directly upon the indicator card can be manufactured. The mathematical relations required are quite simple, and there ought to be no difficulty in constructing an accurate and simple apparatus for the purpose. Of course such an indicator will not draw correct diagrams unless the clearance of the engine has been determined and the apparatus adjusted accordingly. If, however, the indicator can be adjusted until the expansion lines are straight, it will be possible to use the apparatus without determining the clearance.

THE AUTHOR. It is gratifying to find that the development of the new methods of analyzing cylinder performance described in the paper and the conclusions reached regarding them should

[1] For the proof see Practical Thermodynamics, McGraw-Hill Book Co., pp. 33 to 35.

be so generally admitted and approved, especially since these methods involve a most radical departure from the beliefs and teachings which have been held for the last fifty or sixty years.

The only serious objection which has been brought out is the effect of valve leakage, i.e., the leakage of steam through the valve in single-valve engines which does not enter the cylinder, on the method of determining steam consumption.

The relative importance of this objection resolves itself into the actual amount that this form of leakage assumes in practice and the relative size and number of single-valve engines in service as compared with the total horsepower of steam engines used in plants where the highest economy is essential.

The relations of the values of n and the quality of steam at cut-off, given in Fig. 6, are primarily applicable to engines of the general type of that tested, that is, Corliss or four-valve engines in which all the steam passes through the cylinder.

The elimination of valve leakage was the motive which led to the use of a Corliss engine to determine the constants given, as explained in Par. 14. It is obvious that any steam lost by valve leakage has no effect upon the diagram and therefore cannot be accounted for. The importance of leakage of any kind in affecting the results of the steam consumption method is fully pointed out in Par. 67, 69, 73 and 74.

The steam consumption method is however a direct means of measuring valve leakage, since the steam and water present in the cylinder may be accurately accounted for. This method has since been used with a high degree of accuracy in connection with the measurement of boiler-feed to measure the valve leakage of a locomotive engine.

The single-valve engine is not an important factor in large steam engine plants where power must be produced at the lowest possible expenditure of steam. It is used more generally in installations where its low first cost is important and where its high steam consumption is not objectionable, as in small isolated plants having need of all exhaust steam for heating, in auxiliary machinery, and also in locomotives and marine engines because their service conditions demand the simplest possible type of engine. However the valves employed in locomotives and marine engines usually leak very little steam, measured as the proportion of the total amount used, on account of the high speed

of the engine giving large specific capacity for the size of the valve, and also because the large size of the valve enables a very close fit to be made.

The greatest field for the steam consumption method will be found in plants where the highest steam economy is essential, or where tests are impracticable due to service conditions, and in this field the small single-valve engine is relatively unimportant.

The engine tested by Mr. George Mitchell and quoted by Mr. Stevens was only 6 in. by 6 in. in size. The 80-h.p. engine tested by Mr. Stevens has a cylinder probably not larger than 12 in. by 12 in. in size. It is only in the very small single-valve engines, which are unimportant from a power standpoint, that valve leakage reaches the value of 20 per cent, a figure Mr. Stevens uses as an average for single-valve engines. Even small single-valve engines may have very little valve leakage as will be seen from the three examples of this type of engine given in Table 4. The two smaller engines, 14½ in. by 13 in. in size, were tested by Mr. George H. Barrus, and the test results together with the diagrams were taken from his book, Steam Engine Tests. The average difference of the steam consumption for these three engines, as determined from the diagram and the test consumption, is 7 per cent, all of them being within the 8 per cent given in the paper.

Mr. Stevens' estimate of the average amount of valve leakage shows that he has totally neglected to take into account the most important type of single-valve engine, the steam locomotive. It is absolutely impossible for the valves of locomotive engines to leak any considerable amount and still obtain the high steam economy which is found on test plants. When a locomotive valve does leak as much as 20 per cent a loud steady "blow" is heard between exhausts.

The effect of leakage upon the application of the steam consumption method may be summed up as follows: If a properly taken indicator diagram is obtained from a noncondensing nonjacketed Corliss or four-valve engine, and if the logarithmic diagram shows that there is no serious leakage, the constants of Fig. 6 will give the steam consumption as stated in Par. 74. If however the same procedure is followed for a single-valve engine of the same type, the result depends upon the amount of valve leakage, but usually with a fair sized engine and the

valves in good condition, the steam consumption may be obtained from the diagram to within a maximum difference of 8 per cent of the test results, as shown in Table 4.

The results obtained by Mr. Rice in his applications of the method corroborate the figures given in the paper.

The variation of the n-X_e law with pressure mentioned by Mr. Moss has been verified further in work on locomotive engines. The distinction between different pressures rests upon the average coördinates of the points found at each pressure, although the average of all pressures gives results which are nearly as close.

Professor Heck's analysis of the heat transfer of series V of Table 1 corroborates the statement in the paper that n is a constant for ordinary conditions. The author has found, as Professor Heck mentions, that there is a tendency at very early cut-off for n to be high at first and then fall to a lower value. Curves No. 3 and 5 in Fig. 30 depart widely from the polytropic law due to the extremely low speed and probably also to steam-valve leakage during the latter part of expansion.

Mr. Marquis describes one of the broadest fields of usefulness for the new analysis, and shows that results may be obtained by its use which have not yet been obtained by means of present methods. A large number of locomotive tests have been analyzed since this paper was written.

Mr. Young questions the statement in Par. 8 concerning the usefulness of older methods of analysis. This statement is fully substantiated by the general results of the analysis. Previously no accurate method of obtaining the steam consumption from the diagram was in use, nor was any method known of determining leakage from the diagram with the engine in regular operation, and although several attempts had been made to determine leakage by the divergence of the expansion line from an equilateral hyperbola, yet the methods followed generally possessed no meaning and gave results of no value. The old method of obtaining the clearance of a steam diagram is fully explained in Par. 136, which shows its unreliability and that it is founded on wrong premises. No method of locating the cyclic events accurately was in use, especially for the events of a locomotive diagram.

The quality of the steam mixture at cut-off, called X_e, obtained

in the engine tested when using superheated steam, is below unity because the degree of superheat at the engine throttle was only from 50 deg. to 125 deg. fahr., and also because of the small 12 in. by 24 in. cylinder. Mr. Young probably has in mind locomotive cylinders of about 27 in. by 28 in., and degrees of superheat of from 200 to 250 deg. fahr. These conditions result in the steam being superheated about 50 to 90 deg. fahr. at cut-off.

The locomotive tests mentioned by Mr. Young have already been analyzed and the data will soon be available for use.

Mr. Stanwood questions the low values of X_c obtained from the engine tested when saturated steam is used and compares them with the figures of Mr. Barrus for the same cut-off values of other engines. The engines mentioned by Mr. Barrus had cylinder displacements several times larger than the small engine tested, and the difference in the value of X_c is due to small size which relatively increases the cylinder surface per cubic foot of displacement space. Mr. Stanwood is right in suspecting that some of these engines were tested by Mr. Barrus, in fact, about half of them were taken from his book, Steam Engine Tests.

The engine used was carefully tested for leakage by the usual standing tests and also by running the engine with steam on the crank end only, and taking off the cylinder head to observe piston leakage. It may be stated that valves and piston were practically tight when judged by these tests. The tightness of these parts was further demonstrated by the form of the logarithmic diagrams obtained from the engine, one of which is shown in Fig. 9.

Professor Ennis' high values of n for compression of steam are probably due to leakage from the steam chest into the cylinder at this point as he states in his discussion. Leakage through the steam valve however does not affect the early part of expansion, as stated by Professor Ennis, but only the latter part of expansion when there is considerable difference in pressure between the steam in the cylinder and that in the steam chest. This phase of leakage is treated in Par. 109, etc.

Professor Cardullo is correct in saying that the law $PV^n=C$ does not perfectly express the adiabatic expansion of steam. This law does express it however to within an average error of less than 1/5 of 1 per cent for saturated steam, as may be seen by referring to Bulletin No. 58, page 81, of the University of Illinois Engineering Experiment Station. However the question

as to whether the polytropic law $PV^n = C$ expresses the adiabatic expansion of steam has no direct bearing on the application of this law to the actual expansions occurring in practice.

The evidence gained from plotting over 1600 logarithmic diagrams from over 60 different engines using steam, air, gas and ammonia amply substantiates the statement that expansion and compression of elastic media obey the polytropic law except where certain exceptional conditions exist.

Professor Cardullo contends that since adiabatic expansion gives a straight line in the logarithmic diagram, it follows that the actual expansion cannot be straight. It must be remembered in this connection that theories are courses of thought which express the causes of facts, and that facts never conform to theories when these theories are incomplete and are based on wrong premises.

The present status of the actual forms of expansion curves in cylinders and the amount and direction of the heat transfer that has therefore taken place to give the curves is as follows: The facts gathered from 1600 examples prove that expansion and compression curves obey substantially the polytropic law under normal conditions of cylinder tightness, etc.; therefore it follows that any theories which lead to other conclusions are incomplete and are not supported by the facts, and hence these theories must be discarded or revised to express the facts.

The bend in the expansion line of Fig. 31 is the same as that caused by large leakage from the cylinder; tight engines do not give this abnormal effect.

Professor Cardullo's hypothesis concerning the relation between the volume and the clearance surface is fully answered by actual examples given in Table 4. Analysis Nos. 106 *a* and 110 are two examples which prove that, given the same values of X_c, the values of n which result are practically the same regardless of cylinder size or form and of engine speed. In this matter, as in the matter of polytropic expansion, facts take precedence over theories.

When steam meters are used to measure the rate of steam flow to reciprocating engines, the makers of this apparatus state that the calibration obtained with steady steam flow no longer holds for the intermittent flow of steam engines, and that these meters have to be recalibrated by an actual test for each individual location to give reliable readings.

The steam consumption of the high and low-pressure cylinders of compound engines check each other very well in work done since this paper was written.

The varying temperatures in a cylinder due to swinging loads do not alter seriously the accuracy of the steam consumption method.

The clearance method offered by Professor Cardullo is the reverse of the old method of constructing graphically a curve of the form $PV^n=C$. It is shorter than the logarithmic method, but it is neither as accurate nor as simple. It is not as accurate for the reasons given in Par. 90, nor as simple because the trigonometric relations used involve a higher degree of mathematical education than the mere selecting of the straight line position on the logarithmic diagram according to directions.

With regard to a further amplification of the analysis, suggested by several of the members who have discussed the paper, the purpose in writing the paper was to establish the broad principles of the analysis, and to give sufficient examples to show how these principles are applied and some of the results attained by their application.

STRENGTH OF STEEL TUBES, PIPES AND CYLINDERS UNDER INTERNAL FLUID PRESSURE

By Reid T. Stewart, Published in The Journal for April

ABSTRACT OF PAPER

A comparison of the theoretical formulae for the strength of steel tubes, pipes, and cylinders under internal fluid pressure shows that:

a Clavarino's formula is theoretically correct for cylinders with attached heads, and Birnie's formula for heads held independently. These formulae are practically applicable to certain classes of seamless steel tubes and cylinders and to critical examination of ordinary commercial steel tubes, pipes and cylinders when sufficiently accurate data are available.

b In commercial welded pipe the variation in thickness of wall, perfection of weld, etc., account for variation in bursting strength of sufficient magnitude to render unnecessary any consideration of Clavarino's or Birnie's condition of head support. All such variations give rise to errors on the side of danger when applying these formulae.

c For ordinary commercial wrought pipe Barlow's formula is to be preferred. This formula is

$$\frac{p}{f}=2\frac{t}{D};\quad p=2f\frac{t}{D};\quad t=\tfrac{1}{2}D\frac{p}{f};\quad f=\tfrac{1}{2}D\frac{p}{t}$$

where D = outside diameter in in.
t = nominal or average thickness of wall in in.
p = internal fluid pressure in lb. per sq. in.
n = safety factor as based on ultimate strength
f = fiber stress in lb. per sq. in.
$= \frac{40{,}000}{n}$ for butt-welded steel pipe
$= \frac{50{,}000}{n}$ for lap-welded steel pipe
$= \frac{60{,}000}{n}$ for seamless steel tubes
$= \frac{28{,}000}{n}$ for wrought-iron pipe

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DISCUSSION

C. M. SAMES.[1] About four years ago the writer had occasion to make a considerable number of calculations on the strength of cylinders, and employed C. Bach's formula given in Des Ingenieurs Taschenbuch "Huette." This formula, adopting Professor Stewart's notation, is identical with Clavarino's, excepting that both numerator and denominator of the fraction under the radical sign are divided by ten. Inspection of a numerical table accompanying Bach's formula revealed the fact that it was possible to write an extremely simple expression that would give practically the same values. This expression is

$$t = \frac{0.42\ pD_2}{f - p}$$

and it yields results in no case deviating more than 1.4 per cent from those obtained by using Bach's (Clavarino's) formula for a range of $f = 3p$ to $f = 100p$. This formula was first published in the 1909 edition of an engineering handbook compiled by the writer. Inasmuch as the value (0.3) assumed for Poisson's ratio is at best an approximation, the simplified expression may be used with confidence where the Clavarino formula is called for.

The same expression may be used in place of the Birnie formula by substituting the numerical coefficient 0.495 for 0.42 when $f = 10p$ to $100p$, and 0.483 when $f = 3p$ to $10p$. In the first case the maximum deviation from the values obtained when using Birnie's formula is less than 1 per cent; in the second, less than 1.6 per cent.

The practical accuracy of this simplified formula may be shown by using it to solve Problem 2 in Professor Stewart's paper. Here it is required to determine the fiber stress f in the wall of a cylinder stressed both longitudinally and circumferentially, the outside diameter being 5.5 in., thickness $t = 0.25$ in. (making $D_2 = 5$ in.), and $p = 1500$ lb. per sq. in. Substituting these values in the simplified formula

$$0.25 = (0.42 \times 1500 \times 5) \div (f - 1500)$$

from which $0.25f - 375 = 3150$, and $f = 14{,}200$ lb. per sq. in. Professor Stewart's method, using his table, gives $f = 14{,}200$ lb.; using the Clavarino formula, $f = 14{,}093$ lb. The discrepancy here is due to the fact that Professor Stewart used the nearest tabular value instead of interpolating.

[1] Editor, Industrial Engineering, New York

T. A. MARSH. Lamé's formula may perhaps be arranged in a more convenient form than that shown in Par. 17, as follows: Using the notations

S = Maximum allowable fiber stress per square inch
R = Outer radius of cylinder, in inches
r = Inner radius of cylinder, in inches
P = Working pressure within the cylinder
$T = R - r$ thickness of the cylinder wall, in inches

$$S = P\frac{R^2+r^2}{R^2-r^2} \quad \ldots\ldots [11]$$

$$R = r\sqrt{\frac{S+P}{S-P}} \quad \ldots\ldots [12]$$

$$r = R\sqrt{\frac{S-P}{S+P}} \quad \ldots\ldots [13]$$

$$P = S\frac{R^2-r^2}{R^2+r^2} \quad \ldots\ldots [14]$$

$$T = r\left(\sqrt{\frac{S+P}{S-P}} - 1\right) \quad \ldots\ldots [15]$$

$$\frac{R}{r} = \sqrt{\frac{S+P}{S-P}} \quad \ldots\ldots [16]$$

Using equation [16] the writer at one time prepared Table 5 of ratios of outer to inner radii over a range of allowable stresses and working pressures such as are usually encountered in the design of hydraulic press cylinders.

In high-pressure work of the nature above referred to, it is evident that metals of low tensile strength have early limitations. Considering equation [16], it is evident that the ratio $\frac{R}{r}$ increases very rapidly as the working pressure approaches the allowable stress S. The ratio becomes infinite when $P = S$ and becomes an imaginary quantity when $P > S$. This means commercially that there is a limiting pressure beyond which it is impossible to design a safe cylinder, and a metal of higher tensile strength must be employed. The cost of materials will define the point beyond which it is economy to resort to better grades of metal.

SANFORD A. MOSS. For the two tests which Professor Stewart gives in Par. 31, and, in fact, for most cases of commercial tubes and

TABLE 5 RATIO OUTSIDE RADII TO INSIDE RADII, THICK CYLINDERS

Allowable Stress of Metal per Sq. In. Section, Lb.	Working Pressure in Cylinder, Lb. per Sq. In.												
	1000	1500	2000	2500	3000	3500	4000	4500	5000	5500	6000	6500	7000
2,000	1.732	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
2,500	1.527	2.000	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
3,000	1.414	1.732	2.236	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
3,500	1.341	1.581	1.915	2.449	.....	.....	.....	.....	.....	.....	.....	.....	.....
4,000	1.291	1.483	1.732	2.081	2.645	.....	.....	.....	.....	.....	.....	.....	.....
4,500	1.253	1.414	1.612	1.871	2.236	2.828	.....	.....	.....	.....	.....	.....	.....
5,000	1.224	1.362	1.527	1.732	2.000	2.380	3.000	.....	.....	.....	.....	.....	.....
5,500	1.201	1.322	1.464	1.633	1.844	2.121	2.516	3.162	.....	.....	.....	.....	.....
6,000	1.183	1.291	1.414	1.558	1.732	1.949	2.236	2.645	3.316	.....	.....	.....	.....
6,500	.....	1.264	1.374	1.500	1.647	1.825	2.049	2.345	2.768	3.464	.....	.....	.....
7,000	.....	1.243	1.341	1.453	1.581	1.732	1.914	2.144	2.449	2.886	3.605	.....	.....
7,500	.....	1.224	1.314	1.414	1.527	1.658	1.813	2.000	2.236	2.549	3.000	3.741	.....
8,000	.....	1.209	1.291	1.381	1.483	1.599	1.732	1.889	2.081	2.323	2.645	3.109	3.872
8,500	.....	1.194	1.271	1.354	1.446	1.548	1.666	1.802	1.963	2.160	2.408	2.738	3.214
9,000	.....	1.183	1.253	1.330	1.414	1.507	1.612	1.732	1.871	2.035	2.236	2.440	2.828
9,500	.....	.....	1.235	1.306	1.386	1.472	1.566	1.673	1.795	1.936	2.104	2.309	2.569
10,000	.....	.....	1.224	1.291	1.362	1.441	1.527	1.623	1.732	1.856	2.000	2.171	2.380
10,500	.....	.....	1.212	1.274	1.341	1.414	1.493	1.581	1.678	1.789	1.915	2.061	2.236
11,000	.....	.....	1.201	1.260	1.322	1.390	1.464	1.544	1.633	1.732	1.844	1.972	2.121
11,500	.....	.....	1.193	1.247	1.306	1.369	1.437	1.511	1.593	1.683	1.784	1.897	2.027
12,000	.....	.....	1.183	1.235	1.291	1.359	1.414	1.483	1.558	1.640	1.732	1.834	1.949
12,500	.....	.....	.....	1.224	1.277	1.333	1.393	1.457	1.527	1.603	1.687	1.779	1.878
13,000	.....	.....	.....	1.215	1.264	1.318	1.374	1.434	1.500	1.570	1.647	1.732	1.825
13,500	.....	.....	.....	1.206	1.253	1.303	1.357	1.414	1.475	1.541	1.612	1.690	1.775
14,000	.....	.....	.....	1.197	1.243	1.291	1.341	1.395	1.453	1.514	1.581	1.653	1.732
14,500	.....	.....	.....	1.189	1.233	1.279	1.327	1.378	1.432	1.490	1.553	1.620	1.693
15,000	.....	.....	.....	1.183	1.224	1.268	1.314	1.362	1.414	1.469	1.527	1.590	1.658
15,500	.....	.....	.....	1.177	1.216	1.258	1.304	1.348	1.397	1.449	1.504	1.563	1.627
16,000	.....	.....	.....	1.170	1.209	1.249	1.291	1.335	1.381	1.431	1.483	1.538	1.599

pipes, the ratio of wall thickness to outer diameter is so small that there is no appreciable difference between the various formulae mentioned. For such cases, say for $\frac{t}{D_1}$ up to about 0.03, Professor Stewart's use of Barlow's formula as being best for engineering purposes is undoubtedly justifiable for the reasons which he gives.

For the case of thick cylinders where the ratio $\frac{t}{D_1}$ has a large value there is a great deal to be said, however, in addition to the data which Professor Stewart gives. For such cases he seems to imply that Clavarino's or Birnie's formulae are proper ones to use for the respective cases with and without end heads. According to Guest's maximum shear law, which is now usually accepted, these formulae are considerably in error.

In any case in which there are stresses in two or three directions at once, some criterion must be known for computation, from the various stresses and strains, of an effective stress, which corresponds, so far as yielding is concerned, to the stress in a bar under simple tension in a testing machine.

The theory of elasticity as it has existed for years gives complete information regarding the computation of the stresses and strains, and the only matter of question is the criterion for the effective stress. In a thick cylinder subject to internal fluid pressure there is a tangential stress which is a maximum at the inner radius, and a radial stress. At the inner radius, the radial stress is equal to the pressure. If the cylinder has heads there is in addition an axial stress, due to the load of the pressure on the inner circle, uniformly distributed over the area given by the difference between the circles.

An old theory of the matter was the "maximum stress" theory. This stated that yielding would begin when the maximum stress in any direction reached the yield point for simple tension, regardless of the existence of the other stresses.

The Lamé formula, cited by Professor Stewart, correctly gives the maximum stress, which is the tangential stress at the inner diameter. Therefore, according to the maximum stress theory, it is the effective stress which must be considered in design work. This theory, however, seems to be wholly abandoned.

Another old theory is the "maximum strain" theory, and it is on this that Clavarino's and Birnie's formulae are based. This states that yielding begins when the strain or deformation in any direction,

due to the stress in that direction as well as to the lateral contraction due to stress or stresses at right angles, reaches the value corresponding to yielding due to simple tension.

The simple stress which would produce such a strain is called the true stress and is found by adding to the actual stress, called the apparent stress in this theory, three-tenths of any stress at right angles with proper signs. The factor 0.3 is the value taken for Poisson's ratio.

In a thick cylinder the maximum strain is the tangential strain at the inner diameter. The Birnie formula, which is the maximum strain formula for a thick cylinder without heads is, therefore, obtained by adding to the tangential stress given by the Lamé formula 0.3 of the internal pressure, which is the radial compressive stress.

The Clavarino formula is the maximum strain formula for the case of a cylinder with heads. The axial tension decreases the tangential strain so that the effective stress for a cylinder with heads is less than when there are no heads, according to the maximum strain theory. This is shown by the difference between the Birnie and Clavarino curves in Figs. 3 and 4.

The "maximum shear" theory originated by Guest in England and Mohr in Germany is now usually accepted by engineers who have investigated the matter. This states that yielding begins when the shearing stress reaches a certain value. Hence, when there are stresses in different directions it is necessary to compute that simple stress which produces an equal maximum shear. It can be shown that when there are two stresses with opposite signs the effective simple stress is the numerical sum of the two values. In the present case the maximum effective stress is at the inner diameter and is found by adding to the tangential tensile stress given by Lamé's formula, the radial compressive stress, given by the internal pressure.

The situation is the same whether there are heads or not, since the axial tension added by the heads is less than the tangential tension and makes no difference in the shear due to the tangential tension and radial compression. The equivalent stress thus computed is greater than that computed by Birnie's formula, since the whole radial stress is added instead of 0.3 of it. It is, of course, greater than Clavarino's formula, since nothing is subtracted on account of the axial tension caused by the heads.

Using Professor Stewart's notation, the effective stress as thus computed on the basis of the maximum shear theory is as follows:

$$\frac{P}{f}=2\frac{t}{D_1}\left(1-\frac{t}{D_1}\right)$$

This is the formula which the writer believes should be used for thick tubes and cylinders instead of the formulae cited by Professor Stewart.

It will be noted that this is the same as Barlow's formula, with the factor $\left(1-\frac{t}{D_1}\right)$, giving a decrease of safe pressure. This formula always gives values lower than the old formulae given by Professor Stewart. As stated, the result is the same with and without the heads, whereas the Clavarino formula gives an increase of safe pressure when there are heads.

PROBLEMS IN NATURAL GAS ENGINEERING

By Thos. R. Weymouth, Published in The Journal for May

ABSTRACT OF PAPER

In the production and transportation of natural gas many problems are encountered requiring special applications of engineering principles. It is the purpose of this paper to point out the most important of these and to outline the methods of solution. A brief discussion is given of the properties of natural gas, including a table of analyses of gases produced in the principal gas fields of the United States, together with a formula connecting the heating value of the natural gas with its specific gravity. This is followed by the development of the original formulae for the flow of gas in pipe lines, the power required for compression and the storage capacity of lines, and examples are worked out showing the general method of design of a transmission system.

DISCUSSION

C. N. Cross. There is one locality in the California oil fields which promises to become commercially important in the production of natural gas, viz., the Sunset-Midway district. A single well has for the past two years been producing sufficient gas to supply the entire city of Los Angeles.

That this statement is not an exaggeration will be more readily understood when it is known that the initial rock pressure of the gas in the field is often more than 2000 lb. per sq. in. When these gas pockets are first opened the string of tools is often thrown from the well and over the top of a 90-ft. derrick. Sometimes large quantities of sand and boulders weighing 15 to 20 lb. each are ejected from the mouth of the well. Under such conditions there are very lively times getting the well under control. For lack of transportation facilities this immense quantity of gas is largely being lost.

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TABLE 4 ANALYSES OF NATURAL GAS FROM THE CALIFORNIA OIL FIELDS. PER CENT BY VOLUME REDUCED TO 60 AND 29.82 IN. MERCURY

Location	Year	O_2	CO_2	(C_2H_4) (C_2H_6)	CH_4	N_2	Lower B.t.u. Junker Calorimeter	Lower B.t.u. by Analysis	Lower B.t.u. by Equation	Specific Gravity
Coalinga No. 1....	1911	....	19.20	....	79.0	1.8	667	709	754	0.7460
Coalinga No. 2....	1911	1.60	3.80	....	94.4	...	776	847	903	0.5974
Sunset No. 1.....	1911	....	22.80	10.7	65.9	...	670	762	678	0.8217
Sunset No. 2.....	1911	0.70	7.70	....	87.5	3.6	....	785	857	0.6434
Sunset No. 3.....	1911	1.30	0.60	3.10	92.3	1.7	921	877	918	0.5824
Analyses made by Bureau of Mines and Published in Bureau of Mines Bulletin, No. 19, 1911										
Santa Maria.....	1909	0.2	15.5	20.2*	62.7	1.4	...	884	675	0.8254
Santa Maria.....	1909	0.2	14.1	19.0	64.6	2.1	...	882	709	0.7910
Torry.....	1910	..	6.8	35.6	54.2	3.4	...	1053	695	0.8051
Coalinga.....	1910	....	11.1	....	88.0	0.9	...	789	836	0.6640
McKittrick.....	1909	...	30.4	1.0	66.2	2.4	...	610	638	0.8616
West Los Angeles...	1909	0.1	1.0	2.7	91.0	5.2	...	859	902	0.5978
Sunset.....	1909	...	10.5	....	87.7	1.8	...	787	838	0.6619
Fullerton.....	1909	...	1.7	9.5	86.7	2.1	...	929	876	0.6241
Fullerton.....	1909	...	1.4	10.9	85.8	1.9	...	944	873	0.6271
Kern River.....	1909	...	6.5	8.0	84.3	1.2	...	884	841	0.6594

* The remainder of this column C_2H_4 only.

There was a trace of H_2 in Sunset No. 2, and of H_2S in Coalinga No. 1 and Sunset No. 1.

Noting the absence of California gases from Mr. Weymouth's table of analysis, the writer presents Table 4. The first five analyses were obtained in connection with the writer's work in the oil fields.

The sum total of the constituents of these five samples varies from 99.0 to 99.9 per cent. This irregularity is due to the fact that the measuring burette used could be read only to the nearest 0.2 per cent, and no attempt was made to determine the rarer gases. The percentage of nitrogen was read from the burette

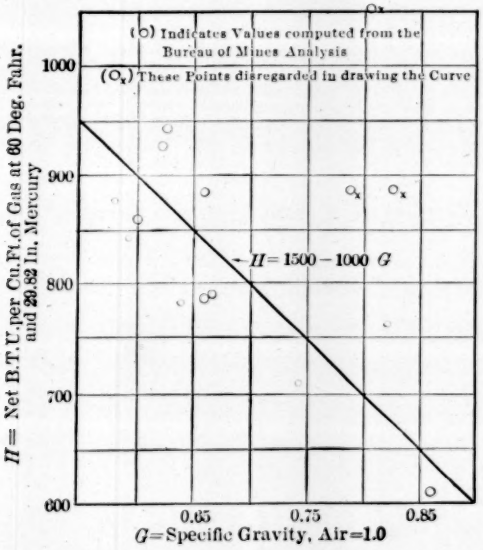
(○) Indicates Values computed from the Bureau of Mines Analysis
(○x) These Points disregarded in drawing the Curve
H = Net B.T.U. per Cu.Ft. of Gas at 60 Deg. Fahr. and 29.92 In. Mercury
1000
900
800
700
600
H = 1500 − 1000 G
0.65
0.75
0.85
G = Specific Gravity, Air = 1.0

FIG. 14 RELATION OF CALORIFIC VALUE AND SPECIFIC GRAVITY FOR CALIFORNIA GASES

instead of subtracting the sum of the other gases from 100 per cent. The remaining ten samples were analyzed by the Bureau of Mines and published in the Bureau of Mines Bulletin, No. 19, 1911. In every case the total of these analyses is 100 per cent.

The peculiar feature of the California gases is the unaccountably high percentages of carbon dioxide present. The fact that California oils are of an asphaltic base may in some way be responsible for the large percentage of carbon dioxide. Roughly, the straight line law connecting the calorific value and the specific gravity, shown by Mr. Weymouth, holds for these gases, but the slope of the line, Fig. 14, is reversed due to the presence of the carbon dioxide in large proportions.

BITUMINOUS COAL PRODUCERS FOR POWER

By C. M. Garland, Published in The Journal for June

ABSTRACT OF PAPER

The paper describes the apparatus and general arrangement of bituminous coal producers as designed for power. The scrubbing apparatus is described in detail, data given on its efficiency, and the amount of solid matter delivered in the gas. Data on the efficiency of the plant, composition of the gas, and operating costs, together with brief discussions on these items are also included. Figures for the first cost and operating costs at full load for a 1200-b.h.p. plant are given in such a way as to make them applicable to different conditions of fuel and load.

DISCUSSION

O. P. Hood expressed the desire to have in this country some way of using very low grade, very high ash fuel. Coal is too cheap now to bring that into immediate use, but that source of supply will doubtless be called upon in future. In looking over Mr. Garland's paper he noticed in the analyses of the coal a great proportion of CH_4. In some of the work at Pittsburgh, in reviewing that particular item, it was found that what has been credited as CH_4 was probably CO, and the analysis of gas which shows a considerable proportion of CH_4 is a little doubtful. The chemists ought to review a little their methods of determining this quantity.

E. D. Dreyfus asked if Mr. Garland in determining his efficiencies determined the power to drive the tar extractor, and how the samples of gas were taken, to which Mr. Garland replied that the figures given in the paper do not include the power for driving the tar extractor, but this power is stated to vary from 5 per cent of the b.h.p. of the *n*. The samples were usually taken

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over a period of about 30 minutes, and drawn through a calibrated meter.

T. A. MARSH asked why, with the published low cost of power from gas producers, public service corporations are buying steam plants. He said he would like to know what can be expected in the large power plant with a peak of 10,000 kw. or over, and when the small plant of say 1000 kw. can be expected to consider producers, say in New York, Pittsburgh, Cleveland and St. Louis.

E. D. DREYFUS stated in this connection that at the present prevailing price of fuel, a producer plant is commercially limited to sizes under 1000 kw.

C. W. BAKER referred to a passage in Mr. Garland's paper in which it is said that with exhaust boilers for gases discharged from the gas engine, these gases should not be reduced in temperature below 220 deg. fahr., unless a cast-iron boiler is used. Mr. Baker's impression is that these gases will produce no corrosion provided their temperature is not reduced below 212 deg., but if there is any sulphur at all in the fuel, even a cast-iron boiler will not stand it long. Within 24 hours Mr. Baker had seen a good solid block of steel that had been a piston rod eaten into a honeycombed mass.

SOME TESTS ON CARBURETERS

By George W. Munro, Published in The Journal for March

ABSTRACT OF PAPER

The paper outlines the problem of carburation for motor vehicle engines and points out the difficulties met in determining the performance of a carbureter as distinct from the performance of the engine to which it is attached.

Results of 400 economy tests on a single engine served by six different carbureters under identical conditions of speed and load are presented in graphical form. By these it is shown that the performance characteristics are different with the different carbureters and that the carbureter characteristics would be determined if those of the engine on which they are superposed were known. Approximate engine characteristics are developed for comparison by using the best performance obtained with any carbureter.

Attention is called to the economic importance of the carbureter, the desirability of applying more scientific methods in its development, and the necessity of having available a testing plant, the constants of which are well determined.

DISCUSSION

F. S. King stated that in his work, he has used gasifiers, as they give more satisfactory results on throttled governed engines, where close regulation is required. The cooling water from the engine water jacket is circulated round the air passages of the gasifier in which a portion of the air supply is carburetted by the gasolene.

This richly carburetted air is then mixed with diluting air in the proportion required for proper combustion. This method gives satisfactory results, even with low-grade oils, good results being obtained with oils as low as 50 deg. in engines with cylinders 11 in. bore or below. On larger cylinders, liquid fuels do not give as satisfactory results, especially in the lower grades, when used in connection with carbureters.

Mr. King emphasized the dependence of the carbureter ac-

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tion on that of the device to which it is attached, and found it particularly so when used with jump-spark ignition: the compressions vary with the load, and carburetors are blamed for the erratic action of varying load conditions which affect the jump-spark more than the carbureter. Mr. King also wished to know if anyone knew, or had any information as to whether the quality of gas in the California natural gas field was uniform from day to day. He had heard of cases where engines operated in a most satisfactory manner for long spaces of time, and without any change in adjustment of mixtures, or change in the quality of lubricating oil or the amount of oil used, an accumulation would take place in the cylinders, without any apparent reason. The ignition system was in first class condition.

F. H. Vose called attention to the fact that there are other difficulties besides those mentioned by Mr. Munro which should be kept in mind during the execution and analysis of carbureter tests. Thus the gasolene is an exceedingly complex fuel, and moreover the commercial product is somewhat variable to say the least, even when purchased through the same sources. Variations may therefore be due to different fuel characteristics, even though the specific gravity for all samples used be nearly identical. Further it seems somewhat doubtful if each carbureter under test receives the proper treatment when the operator depends entirely upon his general observation of the performance of the engine to achieve the best results. In determining the engine characteristics the manograph would prove of great value. It would show the behavior of the explosive mixtures in the engine cylinder, and also check off the behavior of one cylinder against the others: with certain carbureter adjustments the explosions, or "near explosions," will be found to vary widely in character, while other adjustments will give uniform results. The manograph is not advocated here as a means of determining the horsepower of automobile engines, but as an excellent assistant in studying the engine operation.

From certain observations in carbureter adjustments for road service it seems evident that adjustments made for maximum power while the engine is on the testing frame, or with dynamometers attached to the rear axle, do not give the best results on the road; particularly is this true in hill climbing tests. The dif-

ference noted in this characteristic may indicate a slight difference in other characteristics between the dynamometer tests and road service. In this connection the effect of atmospheric conditions upon carbureter action should not be lost sight of. The relative humidity of the air certainly has a marked effect upon the action of some carbureters.

The inlet manifold from the carbureter to the cylinders has an appreciable influence upon carbureter action, particularly on the behavior of the mixture in the different cylinders.

GEO. A. ORROK pointed out that in most or all of the carbureters Mr. Munro tested the automatic devices for supplying the excess air are of an early type, and most of the later ones have better arrangements for handling that. For a carbureter that must work under a variety of conditions, the auxiliary valve for the air is very important. In running motorboats humidities all the way down from 100 per cent to pretty dry air are obtained, and conditions have to be adjusted to suit the air to get full power out of the boat.

THE AUTHOR. The author agrees with Professor Vose that a manograph, had one been available, would have been of material assistance in the work. As it was, the carbureter adjustments were made with the aid of an indicating tachometer and the prony brake, the aim being to carry the maximum load possible at 1200 r.p.m., and in case a variation of fuel consumption was possible under this condition, to adjust for best economy. This was the adjustment throughout the tests on this carbureter.

The carbureters used were all of widely advertised makes, furnished new by the makers especially for these tests. The same inlet manifold and ignition system were used throughout the work, eliminating these elements as variables so far as possible. As the work extended over a period of several weeks, there were of course atmospheric changes entirely beyond control.

The author regrets that the discussion did not touch on what he regards as the principal point of the paper, namely, that a carbureter, an important machine in power production, has essential characteristics which may be determined and plotted, and whether it is worth while to establish the proper facilities for making comprehensive carbureter tests.

NEW PROCESSES FOR CHILLING AND HARDENING CAST IRON

By Thomas D. West, Published in The Journal for June

ABSTRACT OF PAPER

This paper outlines a series of experiments to determine the effect of different methods of treatment in chilling or hardening cast iron during the process of cooling after pouring the mold.

The first experiments showing how to produce mechanically mottled and white iron inside a gray body led to experiments with chillers used in different ways; and with various other heat-absorbing or hardening media, such as air, charcoal, powdered manganese, cyanide, etc. A study was made of the effectiveness of chillers of different thicknesses and of different metals; of the effect of cooling chillers, etc.

The experiments indicate, among other results, that the accepted idea of chilling occurring entirely while the molten metal is solidifying is wrong; and they show how stronger grades of iron can be used for car wheels, rolls, etc., and still obtain the desired depths of chill in such castings. They also demonstrate the superiority of air cooling over metal chillers.

DISCUSSION

Henry M. Howe.[1] Mr. West found that, if after the outer part of a cast-iron casting has solidified and cooled at a normal rate to somewhat below its eutectic freezing point, thus becoming graphitic, the casting is suddenly withdrawn from the mold and quenched in water at a moment when the interior has already solidified, the inside will be of white iron. The fact that the inside of the casting is white, though the normal slow rate of cooling extended below the eutectic freezing point, shows that it solidified as white iron. Because the inside solidified as white iron, the outside evidently did also. The graphitization of the outside must have occurred in that phase of the

[1] Prof. of Metallurgy, Columbia University, New York.

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cooling by which the gray outside had outrun the white interior at the moment when the quenching occurred.

Because the amount by which the cooling of the outside outran that of the inside must have been very slight, the writer infers that this graphitization occurred in a very narrow range of cooling, and presumably below the eutectic freezing point. The reason why the inside remained white may be either (*a*) that it hurried through the temperature just below the eutectic freezing point too fast to allow its cementite to graphitize; or (*b*) that the graphitization was restrained by the pressure put on the interior by the rapid cooling and contraction of the exterior; or (*c*) both these causes may have coöperated. In this particular case his experiments showed that if the slow cooling had not been thus interrupted the interior would have been very gray.

Thus Mr. West confirmed in a very simple and effective way the evidence of Heyn and Bauer that graphitization chiefly occurs immediately below the eutectic freezing point.

The usual occurrence of white cores inside of gray iron pigs so often reported is probably due to pressure. If the outside of the casting becomes firm and rigid, the separation of graphite inside it immediately sets up pressure, because the graphite is so bulky, and the existence of this pressure tends to prevent the further formation of graphite in the parts further in. In most cases this tendency is resisted by the slower cooling of the interior than of the exterior, slow cooling in itself prolonging the opportunity for the graphitization of the cementite.

Thus there is a struggle between slower cooling in the interior which tends to bring the carbon to the state of graphite, and pressure in the interior which tends to prevent it from assuming the form of graphite. In the majority of cases the slower cooling has the mastery, and the interior of the casting is as graphitic as the outside, but in certain cases the effect of pressure has the mastery and restrains the formation of graphite in the interior of the casting. It may often happen that the throwing of a stream of water on the pigs after their outside has turned gray, but before the inside has, or their removal from the sand in cold weather at this point in their cooling, may, as in Mr. West's experiment, lead to an internal chill.

P. Munnoch.[1] It is interesting to find that the phenomenon of internal chill which has puzzled the foundry for so many years has been explained by Mr. West and found to be capable of reproduction. Although the internally chilled castings one sometimes meets in the foundry have not been produced by quenching in water, there is no doubt but that they are due to some rapid cooling effect just at the period when the metal in the center is susceptible to rapid cooling.

In reference to external chilling, the experiments of Mr. West have demonstrated that the chill structure is produced after the metal has solidified. This corroborates the work of E. A. Custer with castings made in chill molds which show no signs of chilled structure provided the castings are removed from the molds immediately after casting.

The placing of a chill in a mold no doubt has a great effect at the beginning of the chilling period, but as the temperature of the face of the chill block becomes heated the chilling effect is reduced considerably before the end of the chilling period, whereas the effect of air chilling will continue equally efficient during the whole of the chilling period. This no doubt accounts for the better results obtained with air cooling by Mr. West.

Paul Kreuzpointner.[2] Summing up the deductions to be drawn from the detailed results of the paper, there are several methods whereby we can obtain a chilled surface in cast iron, the depth of the chill depending upon three factors: (*a*) conductivity of the chilling medium, (*b*) continuity of contact of the chilling medium, and (*c*) quickness of action of the chilling medium, always provided the chemical composition is favorable to obtaining good results.

When chilling cast iron it is aimed to prevent the changing of the combined carbon into graphitic carbon. The interval of time when this change from combined carbon to graphitic carbon takes place being limited, the success of obtaining a satisfactorily chilled casting depends on the united efficiency of the above three factors; hence the importance of having the proper chilling medium.

If the conductivity of the chilling medium is poor, continuity

[1] American Shoe & Foundry Co., Mahwah, N. J.

[2] Secy. American Foundrymen's Association, Altoona, Pa.

of contact and quickness of action will not help much. The best conductive media will give inferior results if the contact is broken too soon or if it is applied too slowly. Of course through intensity of application of one of these three factors we may be able to modify, to a certain extent, the unfavorable influences in weakness of the other two factors, but then we enter the realm of chance and haphazard work.

Since the conductivity of a chilling medium is dependent upon its ability to carry away from the casting a maximum amount of heat in the shortest possible time it is obvious that if the chilling medium is a metallic substance, that metal ought to be as dense as possible in structure, because in such a metal there exists the closest contact between the particles, which in turn permits the heat to travel quickly through the mass of the particles or crystals. Therefore wrought iron would seem to be the least adapted to act as a chilling medium because of the presence of the cinder which acts as an obstruction to the rapid traveling of the heat from one particle to another. Theoretically tool steel would seem to be the best chilling medium of the metal class because of its freedom from interposing films of slag or graphite between the crystals, hence maximum conductivity. However, in every-day practice this is out of the question and it is only mentioned in connection with the question of conductivity of chilling media. Thus for every-day practical operations we have to fall back upon cast iron or a suitable stream of cold air, where this is feasible. In the end it is always the problem to convey the heat away from the casting quickly enough to prevent change of carbon from one state to the other in order to obtain a desired depth of chill.

James A. Beckett.[1] The immersion test referred to in Pars. 63 and 64, indicating the practicability of continuing chilling after the metal solidifies, brings out a new idea in connection with this subject and opens a new field for investigation.

Long experience has demonstrated that for surface chilling nothing better has been found, cost and wearing qualities considered, than the cast-iron chiller, which is also a fairly good absorber of heat, and holds its shape well.

Surface chilling is almost entirely a matter of heat transference, but the efficiency of any chilling medium may be increased

[1] Hoosick Falls, N. Y.

by the selection of a mixture of irons which have the tendency to a greater increase of combined carbon upon remelting than is found in ordinary foundry irons. Irons of a given analysis made from certain eastern ores have this tendency to a greater degree than most southern irons of the same analysis.

The experiment shown in Par. 29 to determine the effect of warm chillers is in accord with the results of practical experience of long standing. For many years, plow castings have been made by using the chiller for the cope of the mold, in the upper surface of which was formed a receptacle, and into this warm or hot water was poured just before the iron was poured into the mold. The effect of the heat was to reduce the depth of chill in the casting, and by securing uniformity in the temperature of the chiller, to produce a uniform density of grain in the fracture without materially decreasing the strength of the casting. In this way, castings were produced with a close grain and susceptible of a very high polish.

Surface chilling results from the instant withdrawal of heat from the surface of molten iron, and is only in a limited degree proportionate to the thickness of the chiller used, as shown in Par. 31. This is true because the body of the chiller, however thick, will not transfer the heat absorbed by the face of the chiller as rapidly as it is absorbed, hence the resort to water cooling and similar devices. The application of a jet of air to aid in the chilling process is a difficult matter, expensive in the long run, and not likely to come into use except in special cases.

Par. 65 contains a statement of two laws, the second of which is: "Graphitization having once taken place in the crust or body of a hot casting, no sudden cooling can restore the carbon to its original combined form. And only by remelting can it be so transformed as to have a chilled or white iron structure." This law is not as positive in its action as Mr. West assumes, except in the case of gray iron. White iron used in the malleable iron industry, when subsequently annealed until practically all its carbon is in the graphitic form, may be restored to its original state in which practically all its carbon is combined by heating it to a cherry red and quenching it in cold water.

CARL HERING.[1] This paper is a good illustration of the importance of studying more carefully the phenomenon of the flow

[1] Cons. Elec. Engr., 929 Chestnut St., Philadelphia, Pa.

of heat through bodies and to other bodies. Like an electric current, a flow of heat encounters a resistance which opposes or checks it more or less, hence when the flow should be rapid, as in the chilling of a casting, this resistance to its flow should be kept as small as possible. The thermal resistance of the cast metal itself is a physical property of the material and is therefore beyond control, but the resistance between it and the chilling material can be controlled, and it is this resistance which is probably so much greater than all the others that it virtually governs the flow of the heat.

The thermal resistance of a joint between two solid bodies, that is, at the surface of a mechanical (not soldered) contact, seems to be relatively very high. This was shown in the case of fire brick in the experiment quoted by the author in Par. 45 and Fig. 15 from an article of the writer's. But with metals this resistance seems to be relatively far greater, as is shown by the fact that an iron bar with a transverse crack part way through it near its end will, when that end is heated to redness, become red up to the crack, while it is still black on the other side. This shows that the heat in flowing to the cold end meets with a very high thermal resistance at the crack, even though it may be only an exceedingly thin one.

In chilling a casting like a car wheel with an iron chiller the heat must traverse the surface of contact and as the cast metal shrinks and the chiller expands, an actual separation must occur, and the very high thermal resistance introduced thereby will no doubt govern the flow of heat and reduce it greatly. This is shown by Mr. West's experiments in which he obtained a better flow of heat when he applied a pressure between the metal chiller and the shrinking casting. It is evident, however, that such a pressure contact cannot possibly be made between two cylindrical surfaces, as in casting car wheels.

Another and sometimes extremely high thermal resistance exists between a metal and gas.[1] A thin layer of very high resistance seems to be produced on the immediate surface. This is what limits the rate of the flow of heat from the flames to the water tubes of a steam boiler. A postage stamp pasted on the outside of a tin cup in which water is boiled over a flame will not char, showing an enormous drop of temperature through a

[1] Flow of Heat Through Contact Surfaces, Met. & Chem. Engrg., January, 1912, p. 40.

layer of only a few thousandths of an inch in thickness, which is merely another way of stating that this layer has an extremely high thermal resistance.

This high surface resistance can be destroyed mechanically by carrying away this film of gas, and this is what Mr. West seems to have accomplished by chilling by means of a rapid flow of air in immediate contact with the metal. His results seem to confirm this theory, or if the theory is acknowledged to be correct, his method of air chilling is a rational one, since it enables him to destroy this high contact resistance, which it would be difficult to do in practice between two metallic surfaces, and apparently would be quite impossible when the surfaces are cylindrical as in the case of car wheels.

Another advantage in cooling with rapidly flowing air is that it affords a possibility of controlling the flow of heat from the casting, hence also the depth of chill and its uniformity in successive castings. This would not be easy with metal chillers. Moreover iron chillers are likely to be in actual, good, compressed contact at only a few points, and as the thermal resistance at those points would be considerably lower than at the rest of the surface it is likely that the chilling would be greater at those points, hence not uniform.

A. S. Dowler[1] said in a communication that in his many years' experience in the wheel business he had read many articles on processes of chilling and controlling the chill, but none which mastered the subject so completely as this paper. A method which can increase the depth of chilling after a crust has commenced to expand away from the chiller should permit the use of softer and often stronger grades of iron than are ordinarily permissible in the manufacture of car wheels. Present methods demand a high-chilling iron in order to obtain the desired depth of chill, which, as a rule, is an iron likely to give trouble because of its bordering too closely on a chilled or white structure.

Albert Sauveur.[2] Mr. West's statement to the effect that the chilling of cast iron does not occur entirely while the molten iron is solidifying is undoubtedly true, but the writer thinks that the opposite view is not the accepted idea. That chilling may continue after solidification is not only consistent with what we now

[1] St. Louis Car Wheel Company, St. Louis, Mo.

[2] Prof. of Metallurgy and Metallography, Harvard Univ., Cambridge, Mass.

understand to be the mechanism of that phenomenon but is demanded by it. By the chilling of cast iron is meant the retention of its carbon in the combined condition, that is, as the iron-carbide Fe_3C or cementite. This carbide must be prevented from dissociating, that is, from breaking up into iron and graphite ($Fe_3C = 3\ Fe + C$), or, in other words, its "graphitizing" must not be permitted. Cementite, however, represents an unstable condition of carbon while graphite is the stable form of that element. The tendency on the part of cementite to dissociate is the greater the higher the temperature. It is, therefore, maximum during solidification immediately after its formation, but it does not by any means cease once the cast iron has become solid. Indeed it is well known that cementite may be readily decomposed at a temperature as low as 800 deg. cent., that is, some 350 deg. lower than the solidification point of cast iron. To illustrate, cast iron may easily be conceived to solidify so quickly that its carbon is retained entirely in the combined condition, but to cool so slowly afterwards that most, if not all, the carbon passes to the graphitic condition. Clearly the chilling due to rapid solidification was entirely obliterated by subsequent slow cooling. That the chilling, therefore, may continue long after solidification is indeed to be expected.

Bradley Stoughton. Mr. West has again put the foundry industry under obligation for his elucidation of the operation of the chilling of cast iron. It may confidently be expected that his investigations will result in improving the quality of chilled car wheels, cast-iron rolls, plows and other objects made of chilled cast iron.

In two respects the writer must differ with the author of the paper: first, with regard to his statement in Par. 65, "Graphitization having once taken place in the crust or body of a hot casting, no sudden cooling can restore the carbon to its original combined form, and only by remelting can it be so transformed as to have a chilled or white iron structure." While this statement is no doubt correct under the conditions of his experiments, and in the sense in which he probably intended it, it cannot properly stand in the terms used in his paper, in view of the positive results of G. B. Upton[1] to the contrary, and, in a minor degree, in view of the well-known whitening of some gray and

[1] The Journal of Physical Chemistry, October 1908, vol. 12.

annealed cast irons when they are reheated to temperatures far below their melting point and plunged into water. That is to say, reheating and plunging into water may accomplish what mere plunging into water from a high temperature during cooling after solidification will not accomplish, and what Mr. West says can be accomplished only by remelting.

The writer's second difference relates to the author's statement that it is an accepted idea that chilling occurs entirely while the molten metal is solidifying. It may be an accepted idea among foundrymen who are not familiar with experiments of others in this field, but it has been known and published by more than one observer that chilling may be effected by rapid cooling subsequent to solidification. This correction does not in any way disparage the value of Mr. West's work; on the contrary, the accuracy of his experiments is further confirmed by the fact that they have corroborated the work of others without any previous prejudice in that direction on the part of the investigator.

J. E. Johnson, Jr., said that at the Ashland plant of the Lake Superior Iron & Chemical Co., of which he was manager, they had been conducting a research for a year and a half past on the general subject of the quality of the iron, and particularly on the subject of iron with an internal chill. In charcoal iron and also in coke iron they sometimes found a spot the size of a nickel, chilled perfectly white, in the center of a pig which was perfectly gray on the outside. They do not know the cause, but believe it to be the result of segregation. Analysis showed ½ per cent more carbon in the gray exterior than in the chilled center, but this result has not been confirmed in all cases. The current explanation among scientific people that the central portion is prevented from expanding (which must occur when the graphite is evolved), by previous solidification of the crust around it, is not valid as they see it, because all irons are cooled under these conditions, and if one shows an internal chill all should.

They have found that these irons are embraced in what are known as high cleavage irons, which means in plain English, rotten irons of very low strength. This character follows the iron throughout, and in a remelt will show up about the same as in the original iron. Even two or three remeltings will not alter the character of the iron unless something radical is done

to change its nature. It will still be spotted iron. As a result, his company never ship this iron to any user who does not alter the fundamental character of the iron before putting it into service.

Their researches have reached the place where they are certain there is virtually no relation between the fundamental quality of the iron and its composition as determined by analysis. There is a certain relation between the percentage of silicon and its chilling quality, but beyond that, with regard to its fundamental characteristics, such as strength, hardness, endurance, etc., an analysis for all the ordinary elements, including carbon, gives no reliable indication. Physical tests, the fracture of the iron, the surface of the pig, its grain and other points must all be considered at least as much as chemical analysis. Even the microscope does not give results that can be depended upon absolutely. His company has been making microscopic examinations of the various irons that they could obtain for over a year, and no relation between the microstructure and the physical qualities has been discovered that they could tie to.

To illustrate the utter unreliability of chemical analysis, Mr. Johnson quoted the records of two cases which were practically identical in all elements, including carbon. With every cast are poured 1¼-in. round test bars. The bars from one of these casts broke at an average of 2725 lb. and those from the other at 4950 lb., or nearly double as much.

The chilled car wheel is bound to be used for years to come and the subject of chilling irons is of enormous importance. Probably 95 per cent of all the freight traffic of the country is carried on on chilled cast iron car wheels. The steel wheel has not, by any means, fulfilled the promise of its advent. The problem is to make the chilled cast iron car wheel good, and the writer believed that could be done by the use of charcoal iron and by that only. He had definite and positive knowledge of one foundry, operated by a railroad, under the care of a man who knows his business, where they make wheels for their own use exclusively. Instead of buying junk and treating it with ferro-manganese and other medicines, he buys selected brands of charcoal iron, all tested under his own supervision for their strength and chilling qualities. As a result, he gets a life for his wheels, in service, on one of the great trunk lines of the country,

of 95,000 miles, as against a life of from 31,000 to 45,000 for the ordinary wheel made of poorer material.

In concluding, Mr. Johnson read a letter from A. M. Thompson of the Thompson Malleable Iron and Adjustable Clamp Company of Chicago in which he said he had to make some rolls which called for a chilled iron. He made the usual mixture of steel and ordinary iron, and test bars 2 in. by 4 in. in size showed $\frac{5}{8}$ in. chill. Reports from the machine shop and rolling mill indicated that the surface was soft. Sulphur was added to the liquid iron and the rolls still continued soft. Some of them were taken out of the chill as soon as practicable after casting and cooled in water with no perceptible difference in hardening. A charcoal chilling iron was then purchased which proved entirely satisfactory. Previous to that he had believed in common with many foundrymen that where the fracture was white the carbon was combined, no matter what iron was used, and the limit of hardness had been reached.

H. M. Lane reported an incident where a foundryman was pouring stove plate. He used a ladle which had been dampened, after drying it slightly on the surface so that only just enough steam would form to cause the iron to boil. He poured one plate from that ladle and another from a perfectly dry ladle. When the plates were broken the first one was found to be chilled on the inside and the other was not. The steam boiling through the iron had caused some change either physical or chemical which produced an internal chill, with $\frac{1}{32}$ to $\frac{1}{16}$ in. of good gray iron on the outside, the inside being hard and white so it could not be drilled. The same thing often occurs when the bed of the cupola has not been dried out thoroughly. It causes the iron to boil at the start and the first 100 or 200 lb. will show internal white marks.

With regard to the addition of sulphur, which has been mentioned, a patent process has been introduced into this country at least five times involving the use of sulphur in steel making. It comes from Germany and consists in adding iron pyrites to a malleable charge to introduce sulphur into the iron. Two-inch and two and one-half-inch sections are chilled all the way through and when they are treated in the annealing furnace they produce a fair grade of malleable.

THE AUTHOR. Mr. Beckett states that "The application of a jet of air to aid in a chilling process is a difficult matter, which will prove expensive in the long run and is not likely to come into use except in special cases." Time only can tell whether his modification of "except in special cases" will hold. The author has patented processes whereby varying conditions can be met with by the use of suction as well as with air pressure. One of these is described in Par. 59 and illustrated in Figs. 9 and 10, showing that with air treatment simple methods can be used to cool and deepen, or harden, a chill. The illustrations of the paper and specimens exhibited at the meeting forcibly demonstrate the effects of air treatment.

In commenting on Par. 65, which says "Graphitization having once taken place in the crust or hot body of a casting, no sudden cooling can restore the carbon to its original form, and only by remelting can it be so transformed as to have a chill or white iron structure," Mr. Beckett gives valuable information but little known. However, it is to be understood that the author had in mind the possibility of changing the state of the carbon at the time of making a casting. This also applies to one of the points raised by Professor Stoughton.

Mr. Munnoch's discussion states that the author's work in demonstrating that the chill structure is produced after the metal has solidified corroborates the work of E. A. Custer. The paper states that the chilling of iron must be accomplished prior to the formation of graphite, and that it has always been thought that in chilling iron the action ceases the moment the metal has solidified. This does not imply that an iron is not chilled until after the metal has solidified. The point desired to be emphasized is that chilling can be continued up to a period of 20 or 30 seconds after the moment of solidification, and Mr. Munnoch has evidently misunderstood the meaning. The writer believes that Mr. Custer's hypothesis, while having some relation to the above question, is not strictly within the lines of research covered by the paper in question.

TESTS OF CHILLABLE IRONS

By Thos. D. West, Published in The Journal for June

ABSTRACT OF PAPER

The tests given in this paper relate to the relative strength of gray iron and of partly or wholly chilled iron, showing the best combination in chilled castings. Many tests are presented of chillable iron alloyed with vanadium and titanium.

Previous to these tests experiments were made for the purpose of establishing a size of round bars suitable for making tests of chillable irons where it is necessary to have the bars either of a uniform gray structure throughout or capable of being chilled throughout, the metal in each case being poured from the same ladle.

The effect is shown on the results of tests of different methods of locating the bar in testing with regard to the quality or grain of the metal. Attention is called to the advisability of drop tests for cast iron and to the complexity and sensitiveness of chillable iron mixtures, requiring delicacy in mixing, melting, casting and testing.

DISCUSSION

A. E. Outerbridge, Jr.[1] In making comparative tests with powdered alloys in ladles of molten iron the writer always uses a cast-iron stirring rod, as he has found that a very appreciable increase in strength is imparted to cast iron by using a wrought-iron rod. He also has two sets of test molds, including long, thin fluidity strips 10 in. long by ¼ in. thick tapering to a feather edge, and uses two cleaned hand ladles, into which he pours the metal from a bull ladle. Both the treated and untreated samples are stirred with cast-iron rods and both poured at the same moment, so that all conditions are alike. Usually shrinkage test pieces are poured for the purpose of developing any tendency to produce "blow" holes at the junction of light and heavy sections. These methods date back to the time when he began and

[1] Metallurgist, Wm. Sellers & Co., Inc., Phila., Pa.

The American Society of Mechanical Engineers, 29 West 39th Street, New York. All discussion is subject to revision.

carried on long continued experiments on the effect of adding small quantities of powdered ferro-manganese to car wheel iron in the ladles, first published in the Journal of The Franklin Institute March 1888.

C. B. MURRAY.[1] Ever since 1908, when Dr. Moldenke read his paper on the tests of vanadium in cast iron, there has been considerable discussion as to how vanadium and titanium do really affect such iron.

When vanadium first made its appearance, and began to be used in steel, it was thought, for a time, that its action was of the nature of a scavenger or a cleaner of the metal, and that the actual presence of vanadium in the finished steel was not necessary. This idea has, however, been cast one side, and now it is pretty generally believed that vanadium acts in two ways: first as a deoxidizer, the resultant vanadium oxide going into the slag; second, that a part of the vanadium should be left in the steel. Just how this acts is somewhat uncertain, but it seems to intensify the action of carbon and other metalloids in the steel. This same theory has been applied to titanium and seems to be still held.

Mr. West has been very painstaking, and seems to have attained his object of securing two specimens of as nearly as possible the same chemical and physical characteristics. The results of the tests, however, especially with regard to the modulus, do not give much information.

The determinations of small amounts of vanadium and titanium are somewhat difficult operations and require a great deal of care and attention. The chemical determinations of these materials were very carefully made in the laboratory of Crowell & Murray in Cleveland and the writer feels assured that the results are correct.

The writer has attempted to get some data from Tables 5-9, but apparently the most marked effects of the addition of these elements is in the chilled round samples. Just why this is so he is unable to say, but it is hardly a coincidence, since in all cases where the chilled round samples can be compared the iron containing either vanadium or titanium, or both, shows a marked increase in their modulus. Possibly the effect of all the carbon being in the combined state and the fact that Mr. West has ap-

[1] Crowell & Murray, Cleveland, O.

parently secured a more homogenous thoroughly chilled sample in the round piece than in the square pieces may have some bearing on this question.

His conclusions in Par. 37 would seem to indicate that he himself is not thoroughly satisfied with the results of the test, and that perhaps some conditions regarding the temperature of pouring, or the thoroughness of mixing, may have escaped his notice, and that these unsatisfactory results may have come from such conditions.

HENRY SOUTHER. The irregularity of the figures given by the author for the modulus of the iron suggests the possibility of internal strains in the cast-iron specimens. The question arises as to whether or not these strains were relieved by partial annealing. If not, it would seem that these great irregularities may be due to uncertain internal stress.

In reference to the effect of vanadium and titanium, it appears that these alloying elements were added in the ladle and that the metal cooled somewhat because of these additions. An indication of increase of strength was obtained. It is fair to assume that the hotter the metal the slower it will cool in the molds. It is a fact that the slower the cooling, the greater the separation of graphitic carbon and the weaker the specimen. After the addition of vanadium or titanium the metal is cooler, therefore chills more quickly with less separation of graphitic carbon and has greater strength. So far as the writer has noted, Mr. West has not taken this into consideration. It is unfortunate that the iron with and without alloying elements was not poured at the same temperature.

One point the writer would like to see cleared up is the question of modulus. Various figures for the modulus appear in many of the tables. They vary widely one from the other and also from what is known of the modulus of the iron. It would be an addition to the paper could Mr. West supply the means of obtaining these figures; that is, it would be well to show the formula used and the methods taken to measure the modulus.

BRADLEY STOUGHTON. The writer is greatly interested in the effect of titanium on the strength, contraction and chilling of iron, because Mr. West's results confirm some hitherto unpublished tests of his own. Mr. West finds that titanium reduces the contraction

and the tendency to chill, which is corroborated by the writer's observations. In an attempt to prevent chilling altogether, he added first 1.14 per cent of titanium, and then 2.29 per cent; since the latter percentage did not prevent chilling as effectively as the former, he concludes in Par. 31, "that iron could not be prevented from chilling beyond a certain limit by its use." The writer's investigations showed that 2 per cent of the usual titanium alloy was sufficient to convert even a dead white iron, in bars 1½ in. in diameter, into a mottled iron, cooled in sand, and that 3 per cent of the alloy was more effective than 2 per cent. In the quality of contraction, the writer's observations corroborate those of Mr. West; namely, that titanium reduces this property. As to the most important characteristic from the commercial standpoint, strength, Mr. West finds that titanium increases the strength of white cast iron by 27 and 32 per cent respectively. The writer's results show an increase in strength all the way from small amounts up to 43 per cent, depending upon the grade of iron, the amount of titanium added and the temperature of pouring. In this connection it may be interesting to note that the writer's investigations cover also the strength of gray cast irons of a great variety of grades, and that the increase produced by adding titanium varies up to 16 per cent.

PAUL KREUZPOINTNER.[2] In determining the quality of a material of construction for a given purpose three factors have to be taken into consideration: (*a*) The nature of the material, (*b*) the nature of the forces tending to destroy the structure, and (*c*) the proper relation of the methods employed in testing to the properties of the material. Concerning the nature of the material, it is obvious that the mechanical means used in testing must conform to that nature. What is suitable for testing cement is not suitable for testing other materials.

It is not quite so simple to determine the best manner for testing a material in accordance with that in which it is subject to destruction when in service. The forces tending to destroy a structural material may be slow-acting or quick-acting; they may be transverse, bending, crushing or tensile forces or a combination of any complicated by variation of temperatures. Thus, chillable irons are chiefly used where they are subjected to

[2] Secy. Am. Fdymen's Assoc., Altoona, Pa.

crushing and bending stresses; hence the best manner of testing them is by the application of a crushing, bending and drop or pendulum test.

However much the qualities of structural materials of the same class may vary in accordance with the requirements for a given purpose, all metals have certain properties which are the same; that is to say all metals are elastic, ductile, strong in power of resistance. Each and all of these properties vary in degree but not in kind. This being so, it may be concluded that according to the third factor in testing materials, the methods must have uniformity in all such cases where the degree of the property of a metal determines its suitability for a given purpose. Thus, if ductility is the measure of quality of a metal for a given purpose, it is by no means a matter of indifference what size of test section is used nor whether the same section is used every time or not.

Erroneous results will surely be the result of testing whenever there is a departure from a standard uniform test section or a recognized standard width of support simply because it is convenient for the time being to vary the method.

As already said all metals are plastic though they vary in degree of plasticity, and to this is due the fact that all metals flow under pressure. But the degree of flow in a test piece under test is free or is influenced by the length of the test section or the width of supports or the distance of the grips. Hence in judging of the value of the results of a series of tests, as in Mr. West's paper, for determining the quality of a metal for a given purpose, it should be ascertained at the same time whether the methods employed, speed of machine, test section, width of supports, etc., represent a recognized uniform standard.

THE AUTHOR. Comments have been made on the value of test bars. It is not so much that we have no information that can be compiled to guide us as it is the indifference and often inability of many to digest what we have. Many test bars are of little value for making comparisons or obtaining the relative physical qualities of cast iron unless properly made. Owing to the lack of proper methods for this work the writer considers the majority of tests to be questionable.

As an example, compare the methods illustrated in Figs. 1 and 2 where one bar provides for two or three tests, with casting

one bar one day for one test and another bar for a second test another day. Under the latter condition there may be a variation in coke, iron and scrap, method of charging, blast pressure, temper of sands, methods and manner of holding, temperature of metals and speed of pouring, etc. The writer has given a great deal of study to the origination of methods best adapted for securing comparative results in the use of test bars. It is shown in the past methods advanced for making test bars as well as in those for obtaining the results given in the paper.

Mr. Souther comments on the evils of variation in pouring temperatures of metals for making comparative test bars. The writer labored to govern this factor, as far as conditions would permit. However, he believes that what irregularities did exist in this factor was not so great as to affect the results. This can be seen, for example, in the difference of strength of like bars having ferro-manganese in them, and those without it as recorded by the tests No. 55 to 59 when compared with those Nos. 65 to 68.

In response to Mr. Souther's request that the formula be given and the method described for measuring the modulus, the following has been prepared by A. G. Smith, who coöperated with the author, and is referred to under "Credit for Professional Coöperation" in the paper:

The question of modulus of rupture which Mr. Souther raises is quite important. We agree that it does not indicate the actual strength of the iron, and is, in fact, only a conventional figure for the reduction to a common basis of tests on similar but slightly varying bars. For example, bars nominally two inches square may actually vary from 1.95 to 2.05 in. in one of both dimensions. If no allowance is made for this variation, the results may be very misleading. The modulus was calculated according to the following very familiar formulæ:

$$M = \frac{3}{2}\frac{Pl}{bd^2} \quad \text{for rectangular bars.}$$

and

$$M = \frac{3}{2}\frac{Pl}{0.59D^3} \quad \text{for round bars.}$$

Where

M = modulus
P = breaking load
l = length of bar in inches (between supports)

b = breadth of bar in inches
d = depth of bar in inches
D = Diameter of bar in inches

The actual measurements of the bars and the actual breaking loads are given with each table.

It is to be understood that none of the figures refer to modulus of elasticity; only to the so-called modulus of rupture.

COMMERCIAL DICTATING MACHINES

At the meeting of the Society in New York, May 14, 1912, the subject of Commercial Dictating Machines[1] was discussed. A. J. McFaul, of the Allen Advertising Company, New York, and Otto Brushaber, of the Dictaphone Company, New York, treated the mechanical method of recording and reproducing sound; and C. K. Fankhauser, of the American Telegraphone Company, Springfield, Mass., the magnetic method of recording sound.

Mr. McFaul said that the dictating machine did not become an article of commerce until the latter part of 1888 and the beginning of 1889. It was the joint invention of Chichester A. Bell, a cousin of Alexander Graham Bell, and Charles Sumner Tainter, who had been working jointly on the problem of recording and reproducing speech for a number of years prior to its public announcement. The graphophone differed from the phonograph, with which Edison had startled the world some ten years previously, in that it recorded sounds by a process of engraving a continuous spiral groove on a wax cylinder or disk, as distinguished from indenting the sound wave on a sheet of tin foil stretched around the mandrel.

The invention of the Bell and Tainter graphophone provided a means for permanently recording and indefinitely reproducing sounds. The original machine, as offered to the public, had somewhat the appearance of a sewing machine, its power being supplied by a treadle and flywheel. The cylinder on which the sound was recorded consisted of a paper base or core with a thin coating of wax composition on which the record was engraved by means of a steel stylus or engraver, mounted on a diaphragm, feeding across the machine while the cylinder revolved in such a manner that the recording tool engraved a spiral groove on the surface of the cylinder.

The official reporters of debates of the United States Senate

[1] Published in abstract only. Complete report may be consulted in the rooms of the Society.

and the House of Representatives and practically all of the leading court reporters in Washington were among the first to adopt the machine into practical work. Its introduction into the work of reporting debates of the Senate and House was one of its earliest triumphs. By the use of the graphophone the official reporters were enabled to cut their amanuenses staff in half, the amanuenses were able to do twice as much work in the same time, and the result was a saving in expense to the reporter and the ability to finish the day's work within one hour or less after the time of adjournment of the respective chambers.

Improvements were made from time to time, simplifying and perfecting the mechanism and operating, but the underlying principle has always remained the same. The substitution of electric and clock-work motors, provision of much more sensitive and accurate recording and reproducing mechanism, development in the material and quality of the cylinder and simplification of the whole mechanism of the machine gradually followed.

The use of the talking machine as a means of entertainment so caught the public fancy that the manufacturers of both the graphophone and the phonograph were practically forced to devote most of their manufacturing facilities and the time and attention of their organizations to meeting the demand for what has now developed into musical instruments of acknowledged supremacy.

Towards the latter part of 1907 a new model dictation machine was designed in the laboratories of the American Graphophone Company, and about the same time the Edison Company brought out a new model of their machine designed expressly for business use; and the almost simultaneous appearance of these machines, coupled with the ability of the manufacturers to give that branch of the business the attention and exploitation it deserves, and the widespread advertising they are now devoting to the subject, have given the dictation machine a fresh impetus.

The use of dictation machines as substitutes for stenographers in office work has been steadily increasing and today there are many thousands of them in daily use throughout the country. They offer to the man whose time is valuable the advantage of being able to dictate whenever he pleases, as rapidly as he pleases, and with the certainty that what he has dictated can be reduced to writing by any intelligent person.

There is no room for doubt that where intelligent and fair

effort has been applied to dictation machines, saving of time, labor and money have been effected. It has been clearly proved that there is a distinct saving in time and an appreciable increase in efficiency on the part of the dictator of from 25 to 50 per cent; while the productive capacity of the typist under normal conditions is increased by 50 per cent and has been known to run as high as 150 per cent.

In one department of a large railroad system 17 stenographers were employed, turning out an average of 75 letters per day. The introduction of the dictating machine enabled them to reduce their stenographic force to eight typists, each one of whom turned out an average of 144 letters per day, an increase of nearly 100 per cent in the productive capacity of the stenographic force accomplished at a reduction in payroll of over 50 per cent.

Another user of dictating machines has certified that the cost of reproducing dictated and typewritten letters has been reduced from 5½ cents per letter by the stenographic method to 2¼ cents by the dictating machine method. Still another user estimated that each machine saves its own cost in operating expenses every 60 days.

Instances of this sort could be cited indefinitely; but the most striking endorsement ever given is that recently published by the United States Government. The President's Commission on Economy and Efficiency in a report recently issued, gives statistics of a test of dictation machines as compared with the use of the shorthand amanuenses from which it deduces that "if the dictation machine were used in all cases where possible in the executive departments in Washington the saving would be more than $500,000 a year."

Mr. Fankhauser said that the principle upon which the Telegraphone is based is that a mass of tempered steel may be impressed with and will retain magnetic fluxes varying in density and in sign in adjacent portions of its mass.

The action of the wire telegraphone in recording is as follows:

A tempered steel wire is caused to pass at a uniform rate of speed successively before and in contact with the pole of a soft iron electro-magnet designated as the "erasing" magnet, and before and in contact with the pole of another similar electro-magnet designated the "recording" magnet.

Through the winding of the erasing magnet is passed an elec-

tric current of sufficient strength practically to saturate the steel wire magnetically and a portion of this current is caused to pass through the winding of the recording magnet in a direction to give the magnet a tendency to reverse the direction of the magnetic flux in the steel wire in order to bring the wire into a magnetically neutral condition.

The winding of the recording magnet is also included in the circuit of the secondary winding of an induction coil, the primary winding of which is in circuit with a battery and telephone transmitter.

Sound waves as from spoken words, etc., impinging upon the transmitter diaphragm and through its vibration setting up correspondent current waves in the circuit of the secondary winding of the induction coil and recording magnet will cause varying fluxes to traverse the magnetic circuit of the recording magnet, which will cause similar fluxes to be impressed upon successive portions of the steel wire as it passes before the pole of the recording magnet.

In reproducing the record, the erasing current is discontinued and a telephone receiver is introduced into the circuit of the recording magnet winding.

Upon causing the steel wire carrying the magnetic record again to pass before the pole of the recording magnet at the same speed and in the same direction as when making the record the varying fluxes in the wire will set up corresponding fluxes in the magnetic circuit of the recording magnet, thus inducing current waves of similar value in the winding and circuit of the recording magnet and the telephone receiver.

The receiver diaphragm vibrating in unison with these currents will set up sound waves similar to those acting upon the telephone transmitter in making the record.

The machine itself may be located at any distance from the dictator. Whether it be 10 ft. or 10 miles it makes no difference in the matter of dictating with comfort and ease. The dictator is supplied with an extension set containing an electric indicator which at all times gives the position of the wire on the reels and enables him to determine just how much of his recording body he has used and how much is yet available without changing the reels. There are three push buttons which give him absolute control of the machine; one for running forward, one for stopping and one for backing it up. In addition to this he is supplied

with a transmitter and a receiver and with a button or spring which may be located in any convenient place either on the extension set or on the receiver or transmitter, which button operates the erasing circuit of the machine. When this button is pressed so as to make contact, and the machine at the same time runs forward, one set of magnets erases any previous records that may be on the wire. Thus he is thoroughly equipped for dictating, for listening to his own dictation, for erasing and making corrections, and in fact for doing everything except changing the reels if they are filled and new ones required. This must be done by some one near the machine.

The arrangement is such that erasing a previous record is not a separate operation. This is automatically done while new dictation is being put on the same wire, for the wire must pass the erasing magnet before it reaches the recording magnet. It is only necessary to press a button and put the erasing magnet in circuit in order to secure a clean wire for the new dictation.

If an error is made or a correction or change is desired the dictator has only to reverse the machine, back up to the desired point, listen to his dictation through the receiver, and redictate that part which is unsatisfactory, at the same time erasing the former or unsatisfactory dictation.

Thus he is at once equipped for 30 minutes' continuous and uninterrupted dictation, at all times having his "stenographer" under perfect control. If he dictates longer than 30 minutes he is interrupted for only the fraction of a minute while the filled reels are being removed and new ones put in the place of them.

It is also a machine for reporting telephonic conversation, and there are appliances for connecting the machine to telephones in cities at great distances and reporting what is said directly into the machine. It is possible even to record speech when a person is standing 10 ft. away from the transmitter.

Mr. Brushaber said that the first idea of the use of the dictating machine was not so much to increase the efficiency of help, but to decrease it by the employment of cheaper labor. The real economy of the dictation machine comes from the fact that the payroll may be cut down as desired or the efficiency of both dictator and transcriber, as compared with the process of taking and transcribing stenographic notes, be increased.

With the dictating machine method of handling correspond-

ence the man reads the letter, gives it the necessary attention in order that he may reply to it properly, and then dictates the reply to the dictation machine. If he requires something from the files in order to enable him to reply to the letter in a proper manner he dictates a memorandum on the dictation machine, and then lays the letter aside in order that the data may be looked up, and brought to him. He then takes up the next letter and dictates the reply, and so on, until he has finished his mail. The dictation of the reply is done while the subject matter of the correspondence is fresh in mind. The work of the correspondence is despatched with ease and accuracy, and the machine may be used instantly without any delays incident to the movements of a stenographer.

In the case of an express company $200 a month was saved after the installation of the dictating machine. The same amount of work is done with nine machines which formerly required thirteen stenographers. There has been an increase of 45 per cent of the letter writing capacity, making it possible to keep their current claims up to date. Moreover it was found that work was transcribed so accurately that it did not require a rereading on the part of the dictator which was formerly necessary under the stenographic method.

A general discussion followed the presentation of the papers.

S. H. Bunnell said he found it a great annoyance to have to stop dictating, often in the middle of an important sentence, to adjust a new cylinder. Another objection he had to the use of the machine was that it was practically impossible to record figures, there being no way for the transcriber to distinguish between fractions and decimals. He also had difficulty in making corrections. In simple dictation, such as letters, etc., changes were easily made, but in dictating specifications, and similar matter, where there are many clauses here and there to be changed it is a source of considerable annoyance.

Jos. F. Diepenbrock[1] said the success of the machine depended on the good will and patience of the operators; also upon the manner in which the dictator talked into it. He had reduced his office staff from five stenographers to two dictating machine transcribers, and had found it possible to cope with a large amount of mail in a very short time.

[1] Harrison, N. J.

T. C. Martin, Jr.[1] found the machine of special value in connection with form letters where he had been able to save time and expense in inserting special paragraphs. There would be less opposition to overcome if in installing the machine the higher officials were given it first, then the lesser ones and subordinates. He considered the dictating machine as important an asset to office equipment as the telephone.

W. W. Macon spoke of the difficulty of recording sibilants. F and S are particularly difficult letters to catch on the dictating machine just as on the telephone.

H. R. Cobleigh said he has used the dictating machine for five years and was thoroughly satisfied with it.

H. J. K. Porter inquired whether or not the machine could be used to handle the proceedings of a meeting conducted in foreign languages, and C. K. Fankhauser answered him in the affirmative.

[1] Secy., Natl. Elec. Lgt. Assoc., New York.

FOREIGN REVIEW

BRIEF ABSTRACTS OF CURRENT ARTICLES IN FOREIGN PERIODICALS

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FOREIGN REVIEW

The aim of the Foreign Review is to present, within the available space, the main data contained in the article indexed. Where possible, reference is made to English or American publications containing fuller information on the subject treated. Measures are given both in original units and their English equivalents. In many instances, engravings and tables are reproduced. Opinions expressed are those of the reviewer, not of the Society. Articles are classified as *c* comparative; *d* descriptive; *e* experimental; *g* general; *h* historical; *m* mathematical; *p* practical; *t* theoretical. Articles of exceptional merit are rated *A* by the reviewer.

Air Machinery

Resonance Phenomena in the Suction Piping of Compressors and Gas Engines, see **Mechanics.**

Staurohr, Apparatus for Measuring Air Volumes and Air Pressures, see **Heating.**

Firing

MECHANICAL FIRING WITH SPECIAL REGARD TO FIRING WITH LIGNITE BRIQUETTES (*Maschinenfeuerungen unter besonderer Berücksichtigung der Braunkohlenbrikettfeuerung*, Weilandt. *Zeits. für Dampfkessel und Maschinenbetrieb*, March 29 and June 14, 1912. 8 pp., 27 figs. *d*). Description of various types of furnaces, with particular attention to those for firing with lignite briquettes. Some of the types described in the article have been already mentioned in this *Review*.

Heating

INVESTIGATIONS AT THE LABORATORY FOR TESTING HEATING AND VENTILATING INSTALLATIONS OF THE ROYAL TECHNICAL HIGH SCHOOL AT BERLIN, WITH AN APPENDIX ON UTILIZATION OF EXHAUST STEAM (*Forschungsarbeiten der Prüfungsanstalt für Heizungs- u. Lüftungseinrichtungen der Kgl. Technischen Hochschule Berlin nebst einem Anhang über "Abwärmeverwertung,"* K. Brabée. *Gesundheits-Ingenieur*, May 25 and June 8, 1912. 22 pp., 65 figs. *ed A*). Condensed account of the work done lately by the above laboratory, with brief description of the apparatus used and some methods of testing. Of the apparatus the most interesting is the one

shown in Fig. 1, called "Staurohr," and used for measuring air-volumes and air pressures. In conjunction with two other apparatus, volumeter and manometer, the Staurohr may also register the air volume and pressure. It consists of a plain, smoothly turned, cylindrical tube enclosed in another tube, both ending in a conical extension. The inner tube takes care of the total pressure, static and dynamic; the outer, through its fine openings, only of the static, which permits the elimination of it in measuring the velocity of the air. The same apparatus has also been adapted for water measurements.

The laboratory made more than 300 tests of protective coverings, and found that they may reduce the efficiency of the heating apparatus as much as 40 per cent.

The article describes very interesting tests on the influence of increased velocity of air and eddies in air on the giving up of heat by heating apparatus, as well as tests of automatic temperature regulators of hot water and low-pressure steam heating installations. The laboratory made extensive

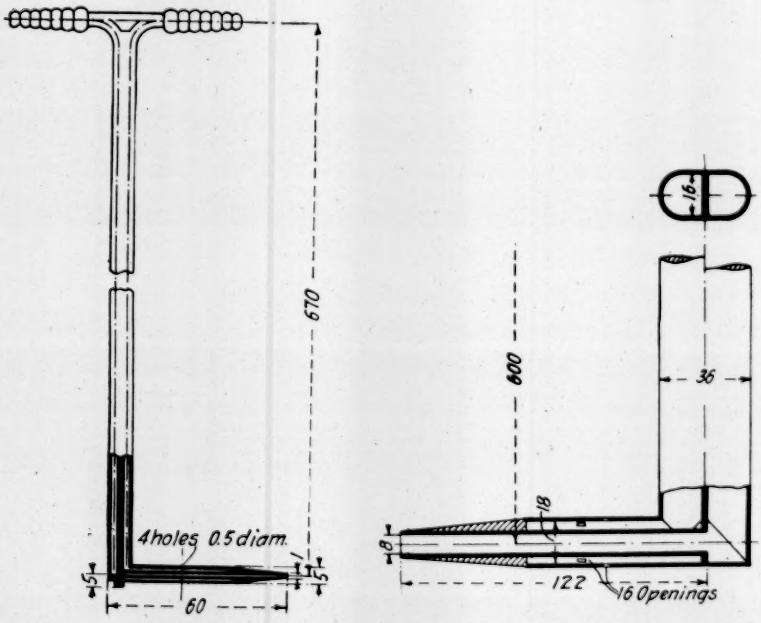

FIG. 1 "STAUROHR" FOR MEASURING AIR VOLUMES AND PRESSURES

tests with steam traps, and found that with large amounts of water the traps work practically with no losses, but show considerable losses with small amounts of water; that the maximum efficiency of the traps depends on the temperature of water, and that the pressures in open outlets cannot always be neglected.

Internal-Combustion Engines

MOTOR FIRE ENGINES (Three articles on the efficiency and construction of motor fire engines under the following titles: *Die Fahrgestelle und Pumpen der Feuerwehr-Motorspritzen*, Schwedtfeger; *Die Entwicklung der Motorspritzen, die Ergebnisse der stattgefundenen und der Wert der geplanten Prüfungsfahrten mit Motorspritzen*, A. Bauschlicher; *Zur Frage der Feuerwehr-Fahrzeuge*, M. Reichel). *Der Motorwagen*, June 10, 1912.

SPRAY CARBURETOR INDICATOR (*Indikator für Einspritz-Vergaser*, G. Bergmann. *Der Motorwagen*, June 20, 1912. 5 pp. 8 figs. *edA*). Beginning of a series of articles on the use of a special device for indicating pressure variations in a high-speed piston engine, the first purpose of the tests having been to investigate the processes in a carburetor of a high-speed benzene motor. A fuller account will be given later.

Resonance Phenomena in the Suction Piping of Compressors and Gas Engines, see **Mechanics.**

Machine Shop

POWER REQUIRED BY BENDING MACHINES (*Der Kraftbedarf von Biegemaschinen*, S. Zander. *Werkstattstechnik*, June 15, 1912. 1 p., 3 figs. *t*). The author shows that the power required for bending machines cannot be determined from the usual formula $M = W. K$, where K is the stress at the yield point, because this formula has been developed on the assumption of the proportionality of deformation to the stress in the fiber, and therefore may be applied only to stresses under the elastic limit, while in the bending machines the stresses which are required to produce a permanent deformation must be carried beyond the elastic limit, to the yield point. The actual power required is considerably higher than would appear from this formula, and may be obtained by substituting for K not the stress at the yield point, but a higher value, such as the breaking strength of the material.

CALCULATION AND DESIGN OF BELT, HEMP ROPE AND WIRE ROPE PULLEYS (*Die Berechnung und Construktion von Riemen-, Hanfseil- und Drahtseil-Scheiben*, Paul Haupt. *Zeits. für Elektrotechnik und Maschinenbau*, June 5, 1912). Beginning of a series of articles on the calculation and design of various pulleys, of a rather elementary character, mainly for young designers.

A SYSTEM OF TAPS AND DIES (*Système de Tarauds et Filières. La Métallurgie*, June 26, 1912. 1/3 p., 4 figs. *d*). Description of a system of taps and dies patented by the Société des ateliers Minne et Stempert and based on the principle of cutting the screw thread by a relieved tool, working only with its back cutting edge, having no side friction, and providing for a free passage of the lubricant down to the cutting edge of the tool, and even under the chip. Fig. 2 A shows how a thread is cut by an ordinary tap filling the space entirely, producing a powerful friction on the thread, and pushing the lubricant in front of the chip. B shows the same kind of work being done by a tap constructed on the above principle, with the faces of the teeth relieved of all unnecessary friction, and plenty of room for the lubricant. Since however in this system the teeth have a pro-

file different from that of the thread to be cut, they have to be disposed in such a manner that their back edges should successively cover the entire face of the thread. C shows how this is done. The thread to be cut is a truncated triangle, and B is a part of the tap; the full lines indicate the profile of the trap, and the broken lines that of the thread. The tap is at first cut with a profile adopted, but sharper than that of the thread, and in a cone of diameter decreasing from a to b in such a manner that the vertices a and b are common to both profiles. Then the exterior of the thread is

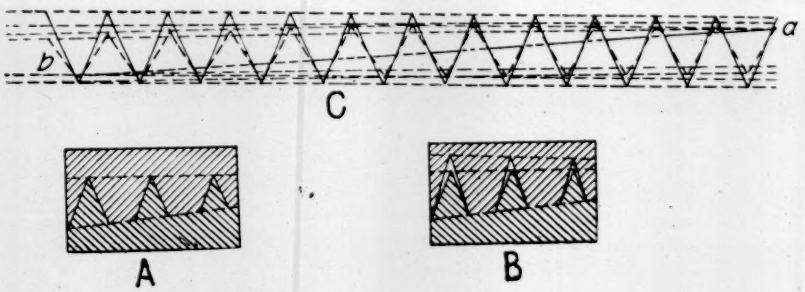

FIG. 2 MINNE AND STEMPERT TAPS

turned in diminishing cones from *b* to *a*, the conical surface obtained being the intersection of the two threads considered. (The above is practically a word for word translation of the original article.)

Mechanics

LOSSES IN THE TRANSMISSION (*Pertes dans la transmission*, G. Lienhard. *La Technique automobile et aérienne*, May 15, 1912. 3½ pp., 4 figs. *t*). Theoretical, mainly mathematical investigation of transmission losses, with special reference to *losses in universal joints*. The author derives the following equations:

For useful work of the joint: $T_u = \frac{\pi}{2} M_o$

For total work: $T = T_u + T_f$

For efficiency of the joint:

$$\rho = \frac{T_u}{T_u + T_f} = 1 - \frac{T_f}{T_u + T_f}$$

where M_o is the moment of resistance at the bearing (say at B, Fig. 3) with respect to the axis of the shaft normal to the plane of the great circle AB, this moment being equal to the force F acting on the journal B. The author shows further that

$$\frac{T_f}{T_u} = \frac{2r \sin \psi}{\pi R} \left[\text{Log } tg \left(\frac{\pi}{4} + \frac{A}{2} \right) + tg\ A \right]$$

and that the efficiency of the joint is zero for angle $A = \frac{\pi}{2}$. A universal joint cannot of course be used for the transmission of efforts at right angles; the efficiency increases as one shaft tends to become the prolongation of the other, and when

they do become so, the efficiency is unity, and the joint is useless. For small values of A, as is generally the case, the value of T_f is small in comparison with T_u, and the equation for efficiency may be written thus:

$$\rho = 1 - \frac{T_f}{T_u}$$

or

$$\rho = 1 - \frac{2r \sin \psi}{\pi R}\left[\log tg\left(\frac{\pi}{4}+\frac{A}{2}\right)+tg\, A\right]$$

Developing it into a series and taking only the first terms:

$$\rho = 1 - \frac{4\, Ar \sin \psi}{R}$$

This form shows clearly the influence of the different elements of the joint. To increase the efficiency: (*a*) it is well to decrease A as far as possible. The

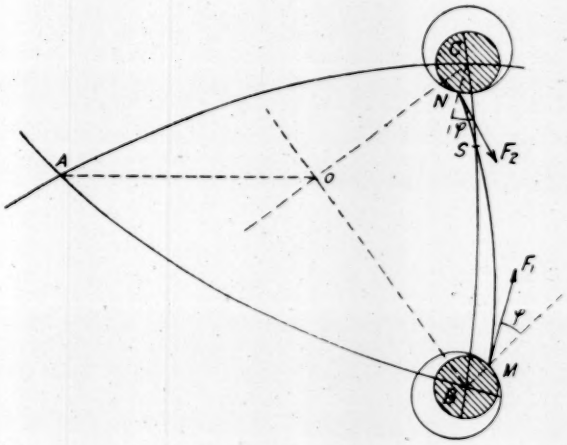

FIG. 3 DIAGRAM OF UNIVERSAL JOINT

author states that although that is practically self-evident, there are still many joints made with the driving and driven shafts, making an appreciable angle between them; (*b*) the radius of the journals must be made as small as possible, and therefore the strongest materials must be used for them; (*c*) the angle ψ must be made as small as possible, or the best lubrication must be used; (*d*) R must be made as large as possible: the influence of this factor is as a rule very little realized.

The author applies his formulae to the examination of a particular joint.

The second part of the article is devoted to the investigation of friction in the steering swivel. The author considers the leading axle, and supposes the load on the axle to be constant; this is really not correct, but the average load differs but very little from the static load on the axle. The author deduces finally the

following equation for the tractive effort ϵ which would produce uniform motion:

$$\epsilon = 2\frac{h}{R}(P+p) + \frac{2\,Pr\sin\psi}{R}$$

where $2P$ is the total load on the axle, $2(P+p)$ the load on the ground, p the weight of a wheel, r radius of the journal, R radius of the wheel, h parameter of resistance to rolling of the wheel which varies with the state of the road and the inflation of the tire, ψ angle of friction at surfaces of contact.[1] This equation shows that the losses in the wheels are the sum of the losses incurred in the journals and in rolling over the ground. It shows also that: (*a*) the weight of the wheel comes in only in one term, and has little influence on the uniformity of motion; (*b*) ϵ decreases as R increases, and there is therefore a considerable advantage in having the radius of the wheel as large as possible; (*c*) h must be as small as possible; it depends directly on the inflation of the tires, and indirectly on the suspension, in that a good suspension decreases h by absorbing the shocks and decreasing the wear of the tires; (*d*) the angle ψ must be made as small as possible. by good lubrication and the use of ball bearings; (*e*) the diameter of the steering swivel must be as small as possible.

UNIVERSAL JOINTS IN AUTOMOBILE AND BOAT CONSTRUCTION (*Kardangelenke im Automobil- und Bootsbau*, K. Regensteiner. *Zeits des Mitteleuropaischen Motorwagen-Vereins*, Mid-June 1912. 10 pp., 13 figs. *tA*). A very interesting investigation, mainly mathematical, of the universal joint, along lines similar to those in the article of G. Lienhardt (p. 1242). Owing to lack of space only the part of the article referring to friction losses in the universal joint is here abstracted.

The friction in a universal joint with a Cardan bolt consists of the friction between the bolt and trunnion block, between block and lateral guide and block and outer guide. The following symbols will be used: G weight of trunnion block in kg.; $m=\frac{G}{g}$ its mass; n speed of driving shaft in revolutions; Md moment of torsion of the driving shaft in kgm.; ρ radius of the housing of universal joint in m.; δ angle between the two axes in arc units; $a\rho$ distance of the center of pressure of a trunnion block in m.; $\beta\rho$ radius of the Cardan bolt in m.; μ coefficient of sliding friction between the housing and block; μ_1 coefficient of sliding friction between nut and block.

The friction between block and nut is due to the action of the force $P=\frac{Md\mu_1}{2a\rho}$. The distance through which the force acts in a second is $s=\frac{4\,\beta\rho.\delta.n}{60}$. The friction loss for two blocks in h.p. is therefore

$$R_{bs}=\frac{4\,Md.\beta\rho.\delta.n\mu_1}{60.75.a\rho}=\frac{4\,Md.\beta\delta.n\mu_1}{60.75.a}=\frac{1}{15.75}Md.\frac{\beta}{a}\delta n\mu_1.$$

The friction between guide and block is due to the action of the force $P_1=\frac{Md.\mu}{2\,a\rho}$, and the distance through which this force acts is $s_1=\frac{4\,a\rho.\delta.n}{60}$, the loss in h.p. being

[1] In the original reference is made to a figure which is not given; Figs. 5 and 6 are omitted in the original.

$$R_{fs}=\frac{4.Md.\alpha\rho.\delta.n.\mu}{60.75.\alpha\rho}=\frac{1}{15.75}.Md.\delta.n.\mu.$$

The friction between the cover of the housing of universal joint and the blocks acted on by centrifugal force is due to the action of a force P_2 proportional to the centrifugal force, and equal to $\frac{m.4\pi^2.n^2.\alpha\rho.\mu}{3000}$ if the center of gravity of the block is $\alpha\rho$ distant from its center, while the distance through which this force acts in a second is $s_2=\frac{4.\rho.\delta.n}{60}$, the loss in h.p. being

$$R_{gs}=\frac{32.m.\pi^2.n^3.\alpha\rho^2.\delta.\mu}{60^3.75}=\frac{320.m.n^3.\alpha\rho^2.\delta.\mu.}{60^3.75}$$

The total loss in h.p. is therefore

$$R=R_{bs}+R_{fs}+R_{gs}=\frac{1}{15.75}Md.\frac{\beta}{\alpha}.\delta.n\mu_1+\frac{1}{15.75}.Md.\delta n.\mu+\frac{320}{60^3.75}mn^3.\alpha\rho^2.\delta\mu=$$
$$\frac{Md.\delta.n}{15.75}\left(\frac{\beta}{\alpha}\mu_1+\mu\right)+\frac{320}{60^3.75}mn^3.\alpha\rho^2.\delta.\mu.$$

In universal joints with two Cardan bolts normal to each other the friction between block and guide and block and housing is eliminated, while the friction between block and nut is twice as large as in the preceding case because the joint has four instead of two blocks. The total friction is therefore

$$R=R_{bs}=\frac{2}{15.75}.Md.\frac{\beta}{\alpha}.\delta.n\mu_1.$$

Usually μ is nearly equal to μ_1, and α is from four to six times as large as β, and on the average may be taken to be five times as large. On this assumption the total loss in the guide may be taken to be five times as large as between bolt and block. The universal joint with two Cardan bolts or with four blocks is in nearly every respect superior to joint of simpler construction.

THEORY OF FRICTION IN LUBRICATED MACHINE PARTS (*Zur Theorie der Reibung geschmierter Maschinenteile*, L. Ubbelohde. *Petroleum*, May 15 and June 5, 1912. 10 pp., 9 figs. *etA*). Continuation of the article on the *theory of friction in lubricated machine parts* abstracted in *The Journal* for June 1912, p. 963. Petroff (*Neue Theorie der Reibung*, Hamburg, 1888), who was the first to investigate theoretically the laws of friction in lubricated bearings, is the originator of the so-called hydrodynamic theory according to which the phenomena of friction in bearings are governed by the laws of internal friction of the lubricant. On the supposition that the lubricant adheres both to the journal and the bearing, and the journal is concentric with its bearing, the velocity fall in the lubricant may be expressed as

$$\frac{dv}{dn}=\frac{U}{\delta}$$

where U is the velocity at the surface of the journal, and δ the thickness of the layer of lubricant supposedly small and uniform. The moment of friction on the journal may then be expressed, according to Newton's law, by the formula

$$M=r.\eta.F\frac{U}{\delta}$$

where r is the radius of the journal, and F the inner wettened surface of the bearing. According to this formula the friction is proportional to the velocity U, contrary to Coulomb's law.

Reynolds and especially Sommerfeld have shown, however, that, contrary to Petroff's assumption, the journal is not concentric with the bearing, but is somewhat eccentric, the eccentricity being proportional to pressure and inversely proportional to velocity. Fig. 4 represents graphically the relation between the velocity U (abscissae) and the coefficient of friction μ. In this figure P is pressure, η viscosity or internal friction of the lubricant, δ difference between the radii of the bearing and journal, which was defined above as thickness of layer of lubricant, because of the supposition of its being uniform. This figure shows that for zero velocity the coefficient of friction has a definite value equal for all pressures; with increasing velocity it decreases a little at first for all pressures, and then begins to increase. The minimum value of the coefficient of friction is the same for all curves (line b), and is equal to $\mu_{min} = 0.943\mu_o$. It depends thus,

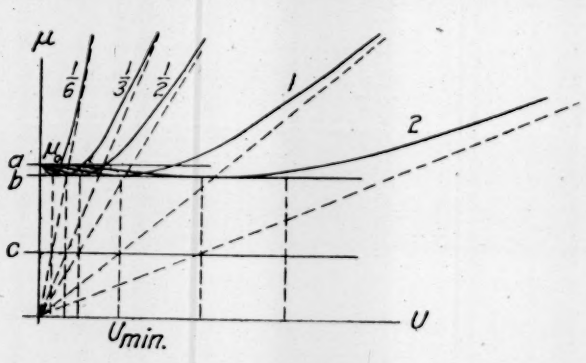

FIG. 4 SOMMERFELD FRICTION DIAGRAM

like the coefficient of friction of rest, only on the dimensions of the bearing, is about 6 per cent less than that coefficient, and is independent of the journal pressure and journal velocity, which are of material influence in shaping the rest of the curve. The greater the pressure the higher is the velocity at which the minimum coefficient of friction is reached. This velocity is denoted by U_{min} and called "velocity of transition," the corresponding pressure P_{min} being called "pressure of transition," and both expressed by the following formulae:

$$U_{min} = \frac{\delta^2 P}{15.1 \eta r^2}$$

and

$$P_{min} = 15.1 \frac{\eta r^2 U}{\delta^2}$$

which show that the value of velocity of transition grows with the growth of the journal pressure P, and with the decrease of viscosity of the lubricant. Since, however, viscosity falls noticeably when temperature rises, we may say that

velocity of transition rises with rise of temperature. At high velocities the separate curves lie asymptotically to broken lines passing through the origin, and representing the equations

$$\mu = \frac{2\pi\eta\, r\, U}{\delta\, P}$$

and

$$\mu = \frac{2\pi\delta}{15.1r} = 0.416\mu_o$$

of which the second is for U_{min}. Accordingly, a line c parallel to the axis of abscissae is drawn at a distance $0.416\mu_o$ from the origin, and the asymptotes of all the curves are obtained by connecting with the origin the points of intersection of this line with the ordinates at U_{min}.

These theoretical propositions have been found to agree fairly well with data obtained experimentally by Striebeck (*Zeits. des Vereines deutscher Ingenieure*, 46, 341, 1902), while the not very considerable discrepancies between the theoretical and experimental values are explained by the author by the fact that, especially at low velocities, we have to deal not with pure liquid friction, but also with dry friction produced by direct contact between journal and bearing (proof-electric contact) which materially increases the total resistance.

To prove the correctness of the contention that as far as the oil is concerned, viscosity is the only factor affecting the coefficient of friction, it was sufficient to show experimentally that all oils of equal viscosity would have the same coefficient of friction in a bearing. Extensive tests in this direction were made by the author with a Martens machine for testing oils, with simultaneous determination of the viscosity of oils tested at the temperature which the oil had during the test (complete table of data is given in the article). These tests have shown that the coefficient of friction is the same not only for all oils having the same viscosity at the same temperature, but even for oils having the same viscosity at different temperature. It was also found that the coefficient of friction increased with the increase in viscosity.

Some of the general results of these tests are of interest. The tests have been made with oils of very different grades, origin, and price, but as far as the coefficient of friction is concerned, practically all the oils were found to behave exactly alike, and particular origin, make, or degree of refinement were found to be immaterial, the only really important mechanical factor being the viscosity. This is true, of course, only when the presence of asphalt or similar matter in the oil is not harmful, as is the case when the oil is not exposed to too high temperatures, but cannot be applied literally in the case of superheated steam cylinder lubrication, lubrication of air compressors, etc., where the oil must be free of easily decomposable substances and otherwise very pure, not for mechanical reasons, however, but because in decomposing it generally forms undesirable residues.

The author does not believe in testing lubricants in the existing special testing machines because: (*a*) the coefficient of friction depends on viscosity alone, and therefore a viscosity determination is entirely sufficient, direct measurement being superfluous; (*b*) it depends quite materially on the shape of the bearing, especially on the difference between the radii of the bearing and journal, and

that the testing machine does not show. It would therefore be entirely wrong to use for practical purposes data obtained on testing machines. It may be pointed out here that Professor Thurston, in his *Treatise on Friction and Lost Work in Machinery and Millwork* (New York, 1885, p. 192), stated that the viscosity test "is regarded as quite as satisfactory as any single physical or chemical test known, and as second only to the best testing machine methods," but at that time he denied (ib. p. 98) that viscosity affects the value of the coefficient of friction.

The author considers one of the greatest difficulties in connection with the problem of efficient lubrication the overcoming of dry friction between the journal and bearing which might theoretically be eliminated only under the following conditions: (*a*) preservation of a definite constant velocity and definite constant pressure; (*b*) definite viscosity of the oil; (*c*) proper dimensions and very smooth surface of the parts in rolling contact. Practically in nearly all cases these conditions cannot be maintained for any length of time. The speed of rotation of machinery changes very frequently; the load naturally varies also; the viscosity of the oil is subject to changes with temperature, while the surface conditions and dimensions of the parts in rolling contact may both vary with temperature, and be affected by wear. Dry friction cannot therefore be eliminated entirely, and the best that can be done is to reduce it as much as possible. The rest of the article is devoted to the discussion of the applicability for this purpose of artificial graphite lubricants, mainly on the basis of data obtained by American investigators.

Note on the Design of Volute Springs (*Contribution au calcul des ressorts à boudin, d'après H. Al. Siebeck. Revue de mécanique*, June 1912. 9 pp., 5 figs., 2 tables. *t*). French translation of the article under the same title in the *Zeits. des Vereines deutscher Ingenieure*, fully abstracted in *The Journal*, April 1912, p. 627.

Resonance Phenomena in the Suction Piping of Compressors and Gas Engines (*Resonanzerscheinungen in der Saugleitung von Kompressoren und Gasmotoren*, P. Voissel. *Zeits. des Vereines deutscher Ingenieure*, May 4, 1912. 3 pp., 11 figs. *etA*). An article by E. W. Koester in *Zeits. des Vereines deutscher Ingenieure*, January 23, 1904, contained a number of indicator diagrams for a Koester compressor. The suction lines in these diagrams showed some regularly occurring oscillations which, when transformed into a time-pressure curve, gave a line similar to a sinusoid, which was the same as long as the speed of the compressor remained constant, but varied in phase with the change of speed. There was a complete oscillation for each stroke. In diagrams taken with the suction piping cut out, the suction line was straight, and coincided with the atmospheric. Similar periodic variations of pressure occur in the piping of gas engines, while their safety of operation requires that the admission of gas and air should be entirely free from irregularities of working, and depend exclusively on the system of governing.

Professors Sommerfeld and Debye were the first to recognize these pressure variations as belonging to the class of resonance phenomena. Since the velocity of the piston in each stroke varies from zero to maximum and

back to zero again, and the air in the cylinder must follow the piston, there arise in the piping longitudinal air oscillations which are reflected at the open end of the pipe. Vertical waves are produced, and the suction piping acts approximately as a "covered" tube, i. e. one closed at one end, and open at the other. In the Sommerfeld-Debye theory a differential equation of the wave motion in the pipe is derived from Euler's fundamental hydrodynamic equation and, by integration, the pressure variation corresponding to given limiting conditions is developed as a periodic time function in the form of a Fourier series. The pressure variation is thus

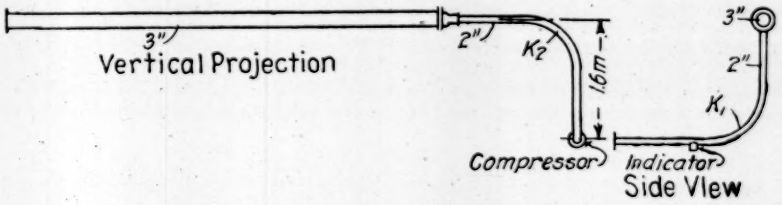

FIG. 5 SUCTION PIPING (ARRANGEMENT FOR TEST)

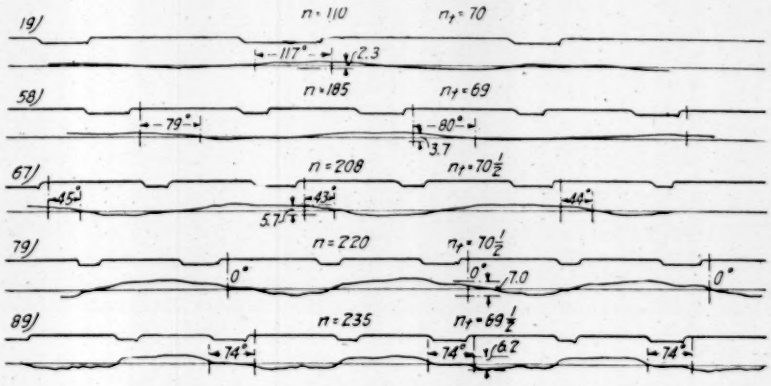

FIG. 6 PRESSURE-SPEED VARIATION IN THE PIPING OF A COMPRESSOR

represented as a converging series of superimposed simple sine waves of increasing frequency, and the resonance can so strengthen each of these waves as to make it predominant or even acting to the exclusion of all others. Friction and inertia dampen the amplitude of vibration of single waves in the case of resonance as well as otherwise, their action being directly proportional to the frequency. Dampening produces also a phase displacement which in case of resonance leads to a lag of a quarter period. The amplitude of single waves is proportional to the compressor stroke and

speed in revolutions. The following equations have been derived for the critical number of revolutions n of the compressor or gas engine (i. e. number of revolutions at which resonance is likely to occur):

For a compressor: $m \frac{n}{60} = (2k + 1) \frac{a}{4l}$

For a gas engine: $m \frac{n}{120} = (2k + 1) \frac{a}{4l}$

Where l is the length of the suction piping, a velocity of sound, k zero, or some integer, and m the number of oscillations of a single wave during one rotation of the crank. For m the following values are given:

For a single-stage compressor: 1, 2, 4, 6, etc.
For a double-stage compressor: 2, 4, 6, etc.
For a single-acting four-stroke cycle engine: 1, 2, 3, 4, 5, etc.
For a double-acting four-stroke cycle engine: 1, 3, 4, 5, etc.

In a double-acting compressor the resonance of each single wave produces rise of pressure at the end of the suction stroke, while in a single-acting compressor the single waves with frequency of one cycle produce rise of pressure only at the critical number of revolutions, and all other waves at resonance. Air pressure variations are especially to be avoided in the case of gas engines.

The article contains a detailed description of the arrangement of tests for the accuracy of the above formulae. Fig. 5 shows the suction piping, 26 m (83 ft.) long, while Fig. 6 shows the pressure variations, which in the case of a single-acting compressor may be represented by

$$-A_1 \cos wt - A_2 \sin 2wt - A_4 \sin 4wt - A_6 \sin 6wt - \ldots\ldots$$

where A is constant, w is the angle velocity of the crank, and t the time. The resonance produces lag in phase of one-quarter period. Opposite each figure are given the actual test data. As to critical velocity the test shows:

Number of Wave	Calculated Velocity	Actual Velocity	Difference
Second	99	110	11
First	198	220	22

In full agreement with theoretical suppositions, there were two oscillations for each revolution of the crank in the second pressure wave, and only one in the first wave. The phase displacement was found to occur just as was expected. At resonance the end of the suction stroke coincided with maximum pressure for the second wave, and with equilibrium pressure in the case of the first wave; the critical number of revolutions according to both the tests and calculations is twice as high for the first wave as for the second. The waves have a regular appearance and have small oscillations only during suction stroke at high speeds, produced by the fluttering of the suction valve.

For complete data of these experiments see *Mitteilungen über Forschungsarbeiten*, Heft 106, p. 27 ff.

A Study of Springs (Étude sur les ressorts, P. Brenier. *Bulletin de la Société de l'Industrie Minérale*, April and May 1912. 103 pp., 28 figs. *tA*). A very interesting theoretical investigation covering the following points: elastic energy

which can be absorbed by a system of solid elastic bodies; flexibility of metal and its strains generally and shearing strains particularly; the application of the former to the study of volute and laminated springs, and the study of laminated springs which can be practically used (the last as an antithesis to the Philipps and Reuleaux equations for the design of laminated springs which, the author claims, are true only for conditions practically never met with in actual design).

Owing to lack of space, only the conclusions, including the main formulae, are given here.

The number of kilogrammeters (=7.216 ft-lb.) absorbed by a kg. of steel under tension or compression stress is equal to $T=\frac{\theta^2}{2E}$ where θ is the stress in kg. per sq. mm. (1 kg. per sq. mm. = 1422 lb. per sq. in.), E coefficient of longitudinal elasticity and T work absorbed per unit volume.

The number of kilogrammeters absorbed per kg. of steel under shearing stress is $T=\frac{\theta^2}{2G}$ where G is the coefficient of rigidity, so that when the stress on the metal is 80 kg. per sq. mm., or say 113,000 lb. per sq. in., the work absorbed by 1 kg. is 51.5 kg-m. or, in American units, the work absorbed per lb. of steel is 167 ft-lb. Only half of this amount may be allowed in case of a volute spring with round spring wires, and only a third in case of laminated springs.

The deflection of a single coil of a volute spring made of round steel wire under a load of 100 kg. (220 lb.) is $f=0.1\frac{D^3}{d^4}$, and the stress under a load P is $t=\frac{8PD}{\pi d^3}$ where D is the mean diameter of the coil, and d the diameter of the wire.

Laminated Springs. A. Study of a given type. L is one-half the length of the developed full-length plate; l one-half the length of the full thickness part of the small plate, or one-half the length of the small plate without staging; n number of plates in spring; q number of plates at the end of spring; b width of plates; h thickness of plates; $2Q$ load applied in the middle of spring; E coefficient of elasticity (20,000 kg. per sq. mm. or 28,500,000 lb. per sq. in.).

Deflection under load $2Q$ is

$$f_{2q}=\frac{12QL^3}{Enbh^3}C \qquad [1]$$

where C is a coefficient depending on $\frac{l}{L}$ and $\frac{q}{n}$, its value being given by a table in the text. If all the plates are not of the same thickness, the thickness of any one of them is taken for h, and then the value n' for each plate of thickness h' is determined from the formula $h'^3=n'h^3$, and the values for n' for all the plates added together. The same is done to determine q. The values in the table were calculated by means of a slide rule, and are exact only to the second decimal place, which is sufficient considering other approximate values used in the equations, and especially the uncertainty as to the value of E. Formula [1] is based on the assumption that the sum of the moments of inertia of all the plates is a linear function of the distance from the middle of the spring in the interval l/L. The article gives a method for determining the deflection when this is not the case.

The stress under the load $2Q$ is

$$t^1 = \frac{6QL}{nbh^2}$$

If all the plates are not of the same thickness, the stress in the p plate is

$$t^1 = \frac{QLe_p}{2\sum_1^n i_p}$$

where e_p is the thickness of the plate (it is well to take the thickest plate), and $\sum_1^n i_p$ is the sum of the moments of inertia of all the plates.

Maximum shearing stress $= \frac{Qi_1}{\sum_1^n i_p}$ and is located at the end of the uppermost plate.

Rational Determination of the Elements of a Laminated Spring. Staging corresponding to the k plate is

$$\frac{L}{\sum_1^n i_p \left(\frac{1}{\rho_p} - \frac{1}{\rho}\right)} i_k \left(\frac{1}{\rho_k} - \frac{1}{\rho}\right)$$

where $\frac{1}{\rho_p}$ is the initial bending of the plate, and $\frac{1}{\rho}$ its bending under maximum load. The successive thicknesses of the plates are determined by the formula

$$e_p \left(\frac{1}{\rho_p} - \frac{1}{\rho}\right) = \frac{2t}{E}$$

where t is the maximum testing stress. The maximum stress allowable in actual service is about one-half of that.

Pumps and Hoisting Machinery

A New Device for Raising Liquids (*Eine neue Fördereinrichtung für Flüssigkeiten*, O. B. *Prometheus*, June 15, 1912. 2 pp., 4 figs. *d*). Description, based on an article in the French periodical *La Nature*, of a pump designed by the French engineer Bessonet-Favre, and consisting of a closely wound spiral with a chain inside (Fig. 7). When the pump which is, as shown in *B*, arranged in the form of an endless chain, travels through a liquid, the latter, through the action of capillary forces as well as direct adhesion, sticks between the coils of the spiral, and the spiral on one hand and the chain on the other, and is carried away with the chain. Theoretically it can be carried to any height or distance as long as it is not subjected to shocks; practically this arrangement has been used successfully in wells 40 m (131.2 ft.) deep. The amount of the liquid carried away by the chain depends of course primarily on the proper dimensioning of the parts. When the chain reaches the top it is carried at a considerable speed over a circular pulley, and the liquid is thrown off by the action of the centrifugal force. The pump is extremely simple, takes very little room, is easily cleaned, easily handled, and is said to be unaffected by cold. It is stated that with a hand-driven pump 3600 l. (say 950 gal.) of water per hour were raised 11 m. (36 ft.). With a double chain of 40 mm. (1.57 in.) in diameter a woman raised 4000 l. (say 1050 gal.) of water 8 m. (26 ft.) in 1 hour. The article does not give more exact data of tests.

VALVELESS PUMP BAVARIA (*Ventillose Pumpe "Bavaria."* *Der Praktische Maschinen-Konstrukteur*, June 6, 1912. 1/2 p., 3 figs. *d*). Description of the valveless pump Bavaria recently placed on the market by the Molkerei-

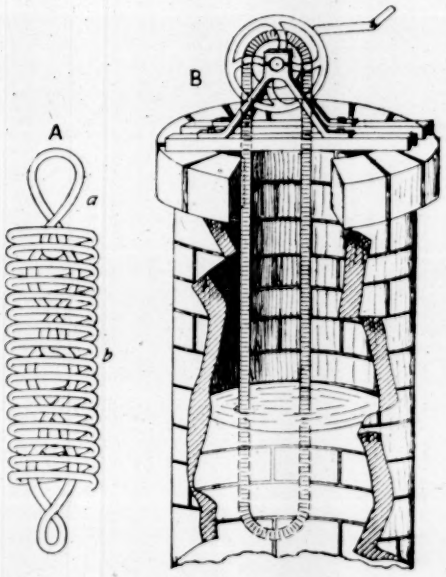

FIG. 7 BESSONET-FAVRE CHAIN PUMP

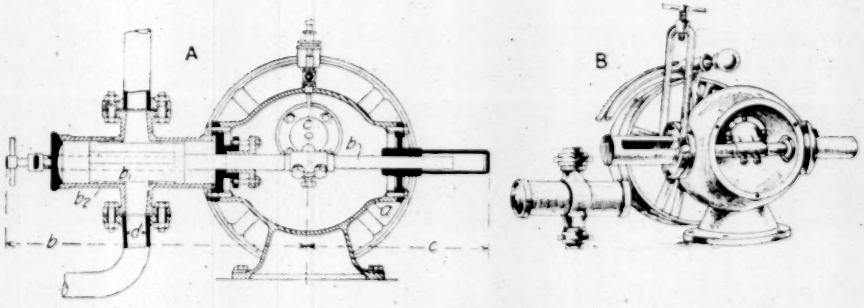

FIG. 8 VALVELESS PUMP BAVARIA

machinenfabrik Gebrüder Bayer in Augsburg, Germany. The peculiarity of the construction of this pump consists in that the suction and compression action is due to the rotation of the piston at *b* (Fig. 8 A), while the place of valves is taken by suitably arranged slots in that piston (B). There are

therefore neither slide valves nor poppet valves, but just a cylinder and a piston. This pump is said to be very convenient for handling thick liquids.

COEFFICIENT OF SAFETY OF WINDING ROPE (*Der Sicherheitsfaktor der Schachtförderseile*, Fr. Herbst. *Glückauf*, June 8, 1912. 11 pp., 6 figs. *t*). The author discusses the question of *coefficient of safety of winding ropes* in mines, mainly with a view to the existing German regulations, and attempts to show that under existing conditions a "coefficient" of safety, i. e. something to multiply by the breaking strength values, is unsatisfactory, and that a "margin of safety," such as was proposed by the Transvaal Commission on Cableways of 1906 is preferable. He points out that with the present (German) system of coefficients of safety the weight of the rope increases much faster than the depth of haulage and load, and that probably more accidents are due to this great increase in the weight of the hauling rope than are prevented by the high coefficients of safety. He recommends therefore the introduction of ropes possessing higher breaking strength, and the admission of lower coefficients of safety for very deep haulage.

TWELVE-STAGES, HIGH-PRESSURE, "INVOLUTE" PUMP (*Zwelfstufige Hochdruck-Evolventenpumpe. Der praktische Maschinen-Konstrukteur*, June 6, 1912. 1/2 p., 3 figs. *d*). Very brief description of some constructive details, but not method of working, of a pump said to be constructed on a new principle of which the exposition is promised to appear in one of the following issues of the paper. An abstract will then be given in this *Review*. This entire issue is devoted to description of pumps and blowers.

Steam Engineering

SIMPLIFIED STEAM ENGINES, M. LUNET SYSTEM (*Machines à vapeur simplifées, système M. Lunet*, M. Lunet. *La Technique moderne*, June 15, 1912. 2 pp., 7 figs. *d*). Description of the Lunet steam engines, single acting and double acting, with steam distribution by means of the piston and fixed expansion. The admission of steam on both sides of the piston is effected at the top by means of an extension of the piston rod and from the bottom by a distributor of a similar kind formed in a thickened part of the same piston rod. The steam escapes through a number of orifices placed horizontally in the middle part of the cylinder, and uncovered by the piston a little before it arrives at the upper and lower dead points. The machine has been tested, but the article does not give any precise data of tests, but the author, who is also its inventor, claims for it a very high efficiency.

STEINMÜLLER WATER SOFTENING APPARATUS (*Épurateur Steinmüller. Electro*, May 1912, 3 pp., 10 figs. *d*). Description of the Steinmüller *water softening apparatus* using the lime and soda process.

USE OF STEAM TURBINES IN MINE HAULAGE PLANTS (*Die Verwendung der Dampfturbinen bei Hauptschachtförderanlagen*, E. Blau. *Dinglers polytechnisches Journal*, June 22, 1912. 5 pp., 9 figs. *d*). Neither the Ilgner system of haulage plants, nor the plants with direct-current motors and

Leonard system of speed regulation proved to be entirely satisfactory and economic. Brown, Boveri & Co. attempted therefore to introduce the turbine as prime mover for haulage plants. The turbines, besides being simple, cheap, absolutely reliable in operation, economic, and perfectly governable, may, if provided with a special automatic by-pass valve, be arranged so as to be able to take care of peak loads. The article describes two types of such valves, both entirely independent of the turbine governor, as well as two types of turbines, one a combined turbine and the other for power production with part of the steam diverted for heating purposes.

THE NEW CONSTRUCTION OF THE NICLAUSSE BOILER (*Die neue Bauart des Niclausse-Kessels*, F. Geiseler. *Zeits. des Vereines deutscher Ingenieure*, May 18, 1912. 4 pp., 18 figs. *d*). The Niclausse boiler is not new in itself, but the article describes a type lately designed to meet the requirements of the French Navy, and containing many important departures from the older type.

As in other water tube boilers, the most frequent cases of bent and broken tubes occurred among the lower tubes which are most strongly exposed to the action of the fire. To obviate this, the following modifications are introduced in the *new type of the Niclausse boiler:* the vertical headers are divided into two parts (Fig. 9 A) by the partition *a* which cuts off the front feedwater passage at about 1/3 of the height of the header (counting from the bottom up). A second partition *b* is installed in the upper drum; it is an extension of the partitions of the headers, extends far over the normal water level nearly up to the beginning of the steam dome, and divides the upper drum longitudinally into two separate parts, with no connection between. The feedwater is introduced into the front part of the upper drum, falls in the form of a rain in the condensing space *c* where owing to the sudden rise of temperature it begins to get rid of its impurities. It then passes into the front port of the header, flows on until it reaches the partition *a*, and is deflected through the inner circulation tubes into the upper nest of boiler tubes, where it is heated up to the required temperature. It is then partly evaporated and freed of all impurities. The free space between the upper and lower nests of tubes is occupied by the superheater. The water having now a temperature of about 200 deg. cent. (392 deg. fahr.) flows through the back port of the header into the back part of the upper drum (A) whence it passes, through the cavity *d* in the partition *b* (see B and D), into the front port of the back header (B) which is not provided with partitions like the headers in A. In large boilers there is one back header, as in B, for each five headers with partitions as in A. The water now feeds all tubes, upper and lower, in B, flows through c_1 into the horizontal connecting chamber *e*, and thence through c_2 into the lower part of the header of A, feeding the lower tubes which were at the start of the circulation cut off by the partition *a*. The water thus flows through all the tubes of the lower rows, and is very easily evaporated since it already reaches these tubes with a temperature of 200 deg. cent. (392 deg. fahr.). C shows a boiler with five headers, with the back header in the middle, the first section of this unit being reproduced in A.

At the tests of the experimental boiler constructed by the Company in 1910 it was found that the lower tubes were nearly absolutely clean while the tubes of the upper rows were full of deposits, but since only the lower tubes are exposed to the fire, while the upper rows are heated by comparatively cold gases, the problem of keeping the tubes exposed to the greatest heat free of deposits was considered to be solved successfully.

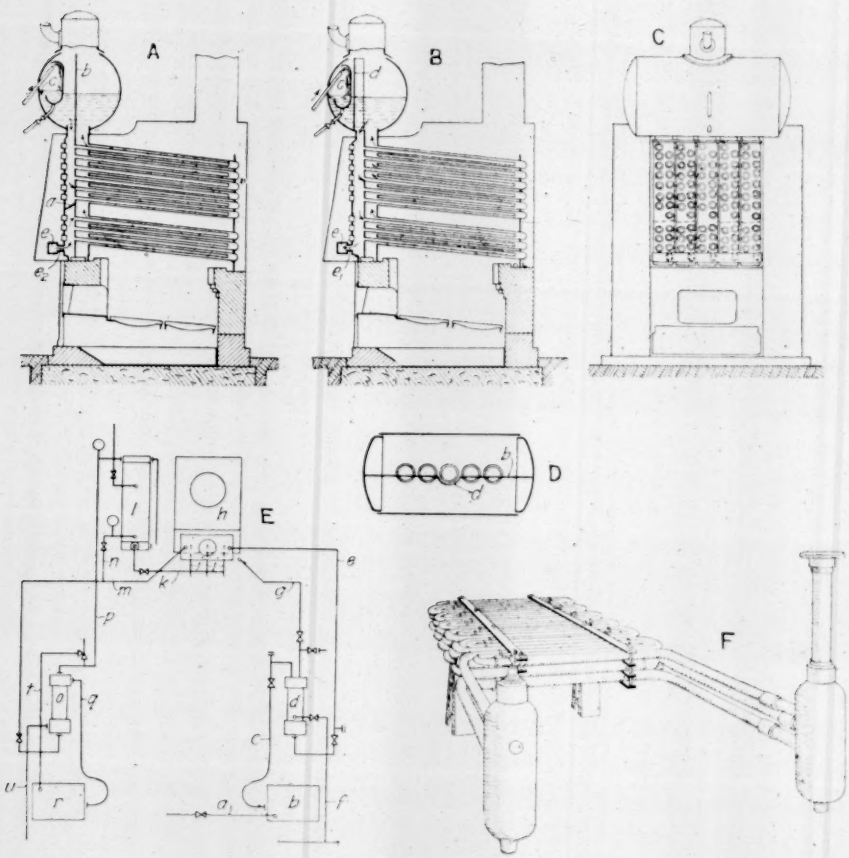

Fig. 9. Niclausse Water Tube Boiler

E shows the arrangement for testing the boilers with oil firing. The purpose of this test was to find out whether the substitution of the back headers for the external downcomers did not interfere with the circulation of water (the tests have shown that it did not), and whether the fire did not destroy the grate too quickly. After a 3½-hour test first with coal, and then oil firing, the boiler was sealed, and next day taken apart: the tubes were found to be clean and straight and the grate in good condition.

F shows the latest design of the Niclausse superheater, such as is inserted between the upper and lower rows of tubes. The holes in the part where there are no tubes are protected by nipples; in some of the boilers with superheaters no holes are made in the middle part of the wrought iron header, so that there are no nipples required. The Niclausse arrangement is claimed to be very convenient because the superheater is protected from the direct action of the fire by the whole lower nest of tubes. Since further it is directly connected with the boiler, and forms practically a single unit with it, it adapts itself automatically to all the fluctuations of service, and is said to make the superheating extremely uniform.

The article describes some auxiliary appliances, such as tube cleaners, brushes, etc.

A Device for Prevention of Flywheel Explosions (*Eine Einrichtung zur Verhütung von Schwungrad-Explosionen*, *Prometheus*, May 25, 1912. 1½ pp., 3 figs. *d*). Description of a *device for the prevention of bursting of a flywheel* owing to excessive speed, manufactured by C. A. Callm in Halle a.d.S., Germany. It is claimed that this device is entirely independent in its operation of both the flywheel and governor; may be adjusted for any desired maximum speed and tested as to its proper working whenever desired by simple means, and that it may be considered as particularly reliable because it has practically no parts which could get jammed at the wrong time, and because it does not begin to act when the flywheel exceeds the set speed, but is in operation all the time that the engine is running. Fig. 10, A and B, show the longitudinal and cross-sections of the apparatus. In the cylinder of the air pump *a* moves up and down the piston *b* driven by the rod *c* from some reciprocating part of the engine. At the side of cylinder *a* is located the valve chest *d*, with an adjustable opening *e* provided with the bolt *m* at the top, and check valve *f* lower down. The air enters through *e* during the forward stroke of the piston *b*, and escapes in the same way during the backward stroke. For starting, the size of the opening *e* is so adjusted that no more air can enter into the cylinder than can escape during the back stroke without compression. When the piston begins to move faster in accordance with the higher speed of the engine, the opening *e* is no longer sufficient for admitting the air, and the check-valve *f* begins to co-operate with it. But this valve closes during the backward stroke of the cylinder, and since only the opening *e* is left to let the air out, the air is compressed in the cylinder. This air pressure acts on the piston *g* loosely held at the end of the cylinder by a spring so that the piston moves forward and, by means of the setting screw *h* and lever *i* depresses the handle *e* and thereby sets free the pawl *k* connected by a wire cable with a throttle valve in the steam supply main. Usually this valve (or an organ corresponding to it) is kept open, but when the pawl *k* becomes free, the valve automatically closes and cuts off the supply of steam to the engine, thus protecting the flywheel from the danger of bursting. The proper working of the apparatus may be tested simply by closing with a finger the opening *e*, which should produce immediately the displacement of the pawl *k*. The apparatus is said to be very sensitive.

INFLUENCE OF CRACKS ON THE STABILITY OF CHIMNEYS (*Der Einfluss von Rissen auf die Standsicherkeit der Schornsteine, Zeits. für Dampfkessel und Maschinenbetrieb*, June 7 and 21, 1912. 6 pp., 13 figs. *p*). Report of a paper on the influence of cracks on the stability of chimneys read by engineer Pietzsch before the International Association of the Societies for Boiler Inspection at a meeting in Konstanz.

The influence of cracks may be at first investigated in the case of a structure of solid square cross-section, that being the simplest case as far as calculation is concerned. In Fig. 11 (A and B) let D be a side of the square, and let the bending moment due to wind pressure produce an eccentric displacement e of the weight of the chimney G within that cross-

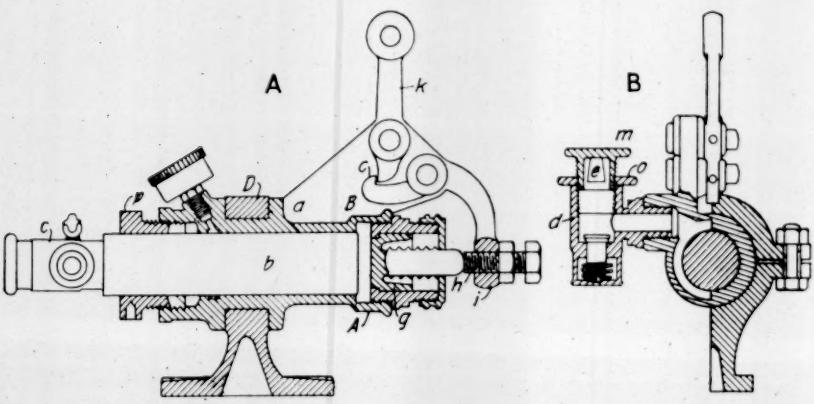

FIG. 10 C. A. CALLM DEVICE FOR PREVENTION OF FLYWHEEL EXPLOSIONS

section. Then, if $c \leq \frac{D}{3}$, the stress diagram will take the form of a wedge with an area equal to G, whence:

$$\frac{3cD\,\sigma_{max}}{2} = G, \text{ and } \sigma_{max} = \frac{2G}{3cD}; z = 3c$$

Here z is the distance of the most compressed fiber from the neutral axis; c the distance of the center of gravity of the wedge from the most stressed edge; the equation $e + c = \frac{D}{2}$ holds good between e and c.

If we assume that the chimney has been cracked in the middle, we will be near the truth in assuming further that each part will henceforward carry one-half of the wind pressure, so that the bending moment due to the wind pressure G will become $\frac{G}{2}e + \frac{G}{2}c$. If then in the case of a chimney free of cracks $c = \frac{D}{3}$ or $c = \frac{D}{6}$, in a chimney with cracks e must still be equal

to $\frac{D}{6}$, and there remains only $\frac{D}{12}$ for c Fig. 12 (C and D). Since, however, for $c = \frac{D}{3}$, $\sigma_{max} = \frac{2G}{D^2}$ for a chimney free of cracks, it becomes, according to Fig. 11 (C and D) for a chimney with cracks $\left(c = \frac{D}{12}\right)$

$$\sigma_{max} = \frac{\frac{2G}{2}}{3D\frac{D}{12}} = \frac{4G}{D^2}$$

i.e., the maximum edge pressure is twice as great as in the case of a chimney without cracks.

If the crack does not appear in the middle of the cross-section, but eccentrically, the same reasoning holds good; the bending moment due to wind pressure is distributed in proportion to the weight of the parts, with the result that the smaller part may be subjected to infinitely large

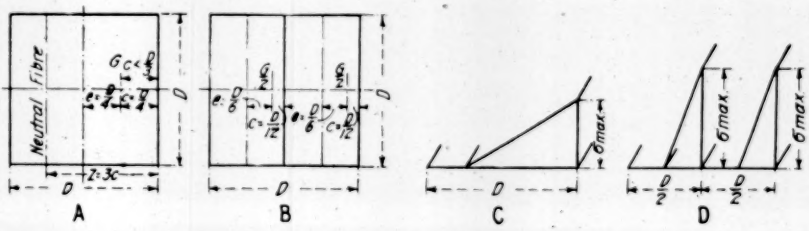

FIG. 11 DISTRIBUTION OF STRESSES IN A CHIMNEY CROSS-SECTION

stresses even while the eccentricity is comparatively small. Practically, however, while the elastic deformation of the smaller part is proportionately greater, the larger part takes care of the greater part of the bending moment.

In principle all these considerations, though actually deduced for a solid rectangular cross-section, are applicable with some modifications to hollow rectangular and annular cross-sections, the actual work of calculation being, of course, considerably more difficult. One thing, however, is certain, and that is that as far as being injured by cracks, the solid rectangular cross-section is the safest, and that chimneys of actual cross-sections are the more endangered by cracks, the more their cross-section differs from the solid rectangle.

Practically the danger from cracks is somewhat less than would appear from the preceding theoretical considerations, because we never have to do with smooth cracks passing clear through the wall of the chimney, and forming smooth surfaces of rupture offering to each other no material frictional resistance. Actually the construction is such that the surfaces of partition interlock, and offer a very considerable resistance to the displacement of separate parts. In most cases therefore it is enough to put an iron ring on a cracked chimney to make it sufficiently safe.

The author quotes the following among causes of the formation of cracks: (*a*) errors in design (weak construction); (*b*) poor workmanship; hasty construction; unfavorable weather during construction, etc.; (*c*) stresses due to thermal expansion; (*d*) weak foundations; (*e*) destruction of the brickwork bond, from outside by weathering, and from inside by the action of acid gases; (*f*) poor materials; (*g*) explosions of gases in the smoke passages. The second part of the article contains data on the governmental regulations for the erection of chimneys in Austria, Prussia, Saxony and Baden, and a comparison of the Austrian and Prussian types.

Strength of Materials and Materials

METHODS OF TESTING OF MATERIALS AND TESTING MACHINES USED BY THE G. DERIHON COMPANY (*Materialprüfungsmethoden und Materialprüfungsmaschinen der G. Derihon-Gesellschaft*, E. Valentin. *Der Motorwagen*, May 20, 1912. 5 pp. 36 figs., and 2 sheets of figs. *d*). The G. Derihon Company, at Loucin, Belgium, used to make forgings for factories of weapons, door locks and railway supplies. Later on it began to make forgings for automobiles, and was so swamped with orders from every part of the world that it had to give up all other work, and besides build a second factory in France. The author ascribes the success of the forgings made by the G. Derihon Company to their methods of testing, which gives a hitherto unsurpassed certainty as to the quality of each piece which leaves the factory. Briefly stated, the Derihon system consists in carefully testing the raw material when received from the factory, and again in manufacture. A test piece is taken from each piece out of which an automobile part is forged, carefully tested, and forwarded to the automobile factory together with the part ordered and data sheet of test. Thus, when sending out an order of 28 steering levers, the company accompanies it with a list of 28 tests; this increases the cost per lever by approximately 5 cents, but gives an absolute certainty that the piece will stand as much as it is expected to. The article describes in detail the organization of the testing department, methods and machines used. (Cp. Automobile, July 4, 1912.)

ON NEW METHODS OF MECHANICAL TESTS OF METALS (*Étude sur les nouvelles méthodes d'essais mécaniques des métaux*, *R. Guillery*. *Revue de métallurgie*, June, 1912. 29 pp. 21 figs. *d*). Description of the author's methods and apparatus for testing the hardness, elastic limit and resilience of metals.

THE VARIATIONS IN THE ACOUSTIC PROPERTIES OF STEEL WITH CHANGES OF TEMPERATURE (*Variations sonores des aciers en fonction de leur température*, Félix Robin. *Revue de métallurgie*, June, 1912. 56 pp., 45 figs. *etA.*) French translation of the work printed under the same title in the *Carnegie Scholarship Memoirs* of the Iron and Steel Institute, vol. 3, 1911, p. 125 ff.

STRENGTH OF ROLLED FLANGED JOINTS AGAINST STRIPPING OFF (*Widerstandsfähigkeit von Flanschenverbindungen die durch Einwalzen befestigt sind, gegen Abstreifen*, Professor Baumann. *Zeits. des Bayerischen Revi-*

sions Vereins, Nos. 7 and 8, 1912, through *Zeits für Dampfkessel und Maschinenbetrieb*, June 7, 1912. *c*). Tests have shown that *rolling* is a sufficiently reliable way of fixing *flanged joints*. The resistance against *stripping off* proved to be so great that the joint was stripped off only at a pressure far exceeding the safe limit for the walls of the pipe. The work of fixing the joint must, however, be done well, as ought always to be the case with piping for steam. The tests have not proved the correctness of the usual assertion that *steel flanges* make a stronger unit with the pipe than *wrought iron flanges*, but have indicated the advisability of using appropriate *tough material for flanges*.

On the Use of Metals in the Construction of Steam Turbines (*De l'emploi des métaux dans la construction des turbines à vapeur*, P. Breuil. *La Technique moderne*, June 1, 1912. 5 pp. *c*). Discussion of the problem of *metals used in turbine construction* for blades and labyrinth packing, containing a synopsis of the information found in technical literature, data obtained from manufacturers, and the author's own considerations. Only data referring to Continental European practice will be quoted here.

For *turbine blades* Zoelly uses steel with 5 per cent nickel. Brown-Boveri, who use a large number of stages, with small falls of pressure, and consequently small velocities of steam, use a special bronze with a coefficient of safety of 15 to 20; the blades do not appear to be subject to wear. The German General Electric Company, which manufactures Curtis turbines, makes the blades out of a special bronze or of steel with a high percentage of nickel, with caulking of a softer metal. Rateau, after unsuccessful experimenting with steel with 25 per cent nickel, uses now steel with 5 per cent nickel, while Rey, of Harlé & Cie. prefers steel with 32 per cent nickel.

Inquiries from various French manufacturers of special steel have shown a great diversity of opinion as to the contents of nickel, but it does not appear to the author that any one of them knows just what influence more or less nickel in the steel will have on the behavior of the metal in a turbine blade. Table (1) gives a resumé of the data which the author collected with respect to bronzes. Of these the composition is given only for the "Durana" bronzes manufactured by the Dürener Metallwerke, Düren, Germany, viz.: the first kind 72 per cent copper and 28 per cent pure zinc, and the second kind 85 per cent copper and 15 per cent manganese. The composition of the Monel metal is given as 70 per cent nickel, 29 per cent copper and 1 per cent iron. Like other bronzes, the Monel metal softens considerably at temperatures above 300 deg. cent. (572 deg. fahr.) and has a relatively low elastic limit, but has the advantage of being little subject to corrosion.

The author calls attention to the fact that on the side of the admission of steam the first blades work in high temperatures, with steam at great velocities and dry; the blades further back are subject to lower temperatures and lower velocities of steam, but the steam is wet, and there is besides the friction of mineral particles carried away by the steam. The guide blades are stationary, and are not subject to the action of centrifugal forces like the rotor blades. It appears therefore that different metals

ought to be used for each of these three classes of blades, but in all cases the metal used must possess great resistance to chemical and mechanical corrosion, and be easily machined by ordinary shop processes. As to chemical corrosion of metals by hot steam there are practically no reliable data. From investigations of the action of salt water on metals it would appear that aluminum and manganese bronzes and Monel metal would give good results, but it is quite possible that chemical corrosion is alto-

TABLE 1 STRENGTH OF METALS USED IN TURBINE CONSTRUCTION

Metal	Elastic Limit		Breaking Strength		Elongation Per Cent
	Kg. per Mm.	Lb. per Sq. In.	Kg. per Mm.	Lb. per Sq. In.	
30 per cent nickel steel	50 to 60	71,000 to 86,000	60 to 80	86,000 to 113,000	40 to 30
5 per cent nickel steel:					
(*a*) annealed	74	105,000	77.5	110,000	36
(*b*) tempered at 800 deg. cent. (1472 deg. fahr.) and then annealed at 600 deg. cent. (1112 deg. fahr.)	90	128,000	104	158,000	9
Brass (67 per cent copper and 33 per cent zinc)	...	...	31	43,000	55 to 60
Brass (70 per cent copper and 30 per cent zinc)	...	...	28	40,000	50
Aluminum bronze BAT2	...	...	55 to 60	78,000 to 86,000	40 to 45
Rubel bronze:					
290 deg. cent. (554 deg. fahr.)	18	25,600	34.19	49,000	43.5
485 deg. cent. (905 deg. fahr.)	13.7	19,500	20.44	29,000	11.9
Durana:					
100 deg. cent. (212 deg. fahr.)	54	77,000	57	81,000	8.5
300 deg. cent. (572 deg. fahr.)	35	50,000	38	54,000	30
400 deg. cent. (752 deg. fahr.)	12	17,000	20	28,500	75
15 per cent manganese bronze:					
100 deg. cent. (212 deg. fahr.)	58	82,500	63	90,000	10.5
300 deg. cent. (572 deg. fahr.)	52	74,000	58	82,500	21
400 deg. cent. (752 deg. fahr.)	33	47,000	38	54,000	68
Monel metal annealed:					
20 deg. cent. (68 deg. fahr.)	24.1	34,000	55.1	78,000	34.5
300 deg. cent. (572 deg. fahr)	19.4	27,500	30.7	43,500	23.5

gether very slight as compared with physical, and the material used ought to be chosen on the basis of resistance to the latter, with respect to which nickel steel appears to have very high qualities with its breaking strength of 60 kg. (80,000 lb. per sq. in.), elastic limit of 40 kg. (57,000 lb. per sq. in.), hardness 180, and elongation 20 to 22 per cent. It is moreover naturally hard, not brittle, and comparatively cheap. There are no data as to its probable behavior at high temperatures, but having a low percentage of nickel it would probably behave like other steel, and would reach the

minimum of its elastic limit and elongation at about 300 deg. cent. (572 deg. fahr.), without, however, becoming brittle. Wickers Maxim make their turbine blades of laminated bars with a steel core and nickel surface, the nickel layer being only a fraction of a millimeter thick.

Miscellanea

STONE BREAKER FOR MACADAM ROADS (Concasseur à macadam, P. B. *Portefeuille économique des machines*, June 1912. 2 pp., 4 figs., and 2 plates of drawings. *d*). Description of the J. B. Aillot *stone breaker* for use in building macadam roads, to be driven by a steam locomobile. It does not break the stone exclusively by crushing but also by slugging, for which purpose the jaws of the breaker are provided with staggered slugger knobs of extra hard and tempered steel. These knobs are not all of the same length, but long knobs on one jaw are placed opposite short knobs on the other. As a result, a stone held against two knobs on one side is broken by the pressure in the middle of the stone by the knob on the other jaw. Further all the knobs are so arranged that the stone, after having been broken by one set of knobs, turns around and falls upon a set of knobs just below, where it is struck in a direction normal to the first. The apparatus is claimed to have given very satisfactory service.

CLEANING OF BLAST FURNACE GASES (*L'épuration des gaz de hauts fourneaux*, C. Herwegh. *Société Industrielle de l'Est*, May 1912. 13 pp., 8 figs. *gd*). A general talk on cleaning blast furnace gases, with an illustrated description of the Feld system.

NON-RETURN VALVE FOR ACETYLENE LIGHTING (*Rückschlagsventil für Acetylenbeleuchtung*, A. J. *Zeits. für Elektrotechnik und Maschinenbau*, June 5, 1912. 1/2 p., 2 figs. *d*). The construction of this valve is based on the same principle as that of the Davy safety lamp, viz. that a flame will not ignite a gas through a fine mesh metal screen, unless the screen itself is sufficiently hot. The valve consists of a casing with two fine mesh screens placed in the path of the gas. The screens are either made of double wire cloth, or of single wire cloth, but with the space between the screens filled with asbestos or some similar substance, which both increases the resistance of the valve to backfiring, and helps mechanically to clean the acetylene. The dimensions of the valve have to be generous enough to permit a free flow of gas, notwithstanding the presence of the screens and auxiliary substances.

EMPLOYMENT BULLETIN

The Society considers it a special obligation and pleasant duty to be the medium of securing better positions for its members. The Secretary gives this his personal attention and is most anxious to receive requests both for positions and for men available. Notices are not repeated except upon special request. Copy for the Bulletin must be in hand before the 12th of the month. The list of men available is made up of members of the Society, and these are on file in the Society office, together with names of other good men not members of the Society, who are capable of filling responsible positions. Information will be sent upon application.

POSITIONS AVAILABLE

0183 Young technical graduate with brains, energy and means wanted, to associate himself in an official position with a company recently formed to develop and extend the sale of automatic machines and other labor-saving devices, under extensive and broad patents. The fields to be operated are the mailing departments of all businesses and wherever commodities are handled in sacks. Shop in Philadelphia.

0184 Technical graduate with from two to five years' practical experience, preferably foundry and machine-shop practice, to fill the position of draftsman and assistant sales engineer for a foundry and machine shop in Montana.

0185 Assistant in department of mechanical engineering of university in middle west, to teach machine design, including steam and gas engines and steam turbine design. Some experience in teaching preference, but not the most essential qualification. Apply through Am. Soc. M. E.

0186 Head of department of mechanical engineering in strong and rapidly growing state college, eastern location. Apply through Am. Soc. M. E.

0187 Young engineer, who has specialized in the design and manufacture of stokers. Salary $2,000. Apply through Am. Soc. M. E.

MEN AVAILABLE

463 Position as manager or superintendent with some reliable company; 23 years' experience in heavy, medium, light and interchangeable work, practical and technical; age 43; married; A 1 references; would like to locate in New York, though location is immaterial.

464 Position as chief engineer, 25 years' experience with mines, smelters and mills in erecting and operating all classes of machinery. Twelve years abroad. No objection to location in any foreign country.

465 Member, 20 years' broad experience in industrial lines, capable of

organizing both shop and selling forces to insure economical operation and of marketing products to best advantage, desires to make change.

466 Motor-truck engineer open for position as transportation expert and efficiency man; competent to investigate and report upon delivery system. Eight years' continuous experience in this industry in designing, constructing, testing motor trucks, and writing on truck subjects.

467 Position desired with mechanical trade paper by editor and writer of several years' experience. Sufficient previous practical training and technical education. Knowledge of cuts, printing, makeup and make-ready, all phases of publishing work. Competent to bear responsibility and direct others.

468 Five-thousand-dollar man desirous of obtaining a position as chief engineer, consulting engineer or shop manager. Wide experience in steam and hydraulic engineering. Unsurpassed references.

469 Young mechanical engineer, technical graduate, wishes position as instructor in mechanical engineering in some middle western or western university or college. Well qualified to fill position. At present employed.

470 Member who has specialized in power transmission efficiency and is especially well posted in the various means of mechanically transmitting power, desires position with manufacturing concern as mechanical engineer or as assistant to the superintendent in charge of engineering. Would consider connection with a consulting engineer. Executive ability, good systematizer. Now holding good position but wants broader opportunities.

471 Member with extensive experience in engine manufacturing plant, 14 years with last company; thorough practical knowledge of follow-up and cost systems and modern shop practice, seeks connection as superintendent or factory manager. Salary not less than $2700-$4000.

472 Technical graduate, 24 years of age, at present employed as assistant to superintendent in large water gas plant, desires to make a change to concern having an opening along mechanical or sales engineering lines. Opportunity and scope desired.

473 Junior, graduate mechanical engineer with five years' selling experience desires position with company manufacturing steam or gas power plant equipment. Location New York or vicinity.

474 Mechanical engineer, age 34, now engaged, valuation work, machinery shops and all classes of buildings, also railroad properties, will be open for assignment.

475 Junior, technical graduate, age 25, three years' experience in building construction and hydraulic dredging, desires position preferably in power plant design or construction work.

476 Works manager, competent to organize all departments of manufacturing plant along modern lines, long experience on light manufacturing, involving interchangeable parts.

477 Member, now employed, desires change, with wide experience in interchangeable manufacturing, especially engines, machine tools and automobiles, can refer to present and former employers; technical education, four years designing, wide experience with machinery and tools, up-to-date on shop organization, management, production, incentive methods of wage payment, and cost reduction. Age 37, married, $3000.

478 Mechanical and electrical engineer, ten years' experience in the design, construction, operation and management of steam and water power plants for industrial and public service companies, desires permanent position with consulting or contracting engineers in or near New York.

ACCESSIONS TO THE LIBRARY

WITH COMMENTS BY THE LIBRARIAN

This list includes only accessions to the library of this Society. Lists of accessions to the libraries of the A. I. E. E. and A. I. M. E. can be secured on request from Calvin W. Rice, Secretary Am. Soc. M. E.

AMERICAN RAILWAY ASSOCIATION. Proc., May 15, 1912. *New York, 1912.* Gift of the association.

BROWN'S DIRECTORY OF AMERICAN GAS COMPANIES, 1912. *New York, 1912.*

BUILDING AND HEALTH LAWS AFFECTING THE CITY OF NEW YORK. Eagle Library no. 121. *Brooklyn, 1912.*

CHICAGO BOARD OF SUPERVISING ENGINEERS CHICAGO TRACTION. 3d Annual Report. *Chicago, 1911.* Gift of Bion J. Arnold.

COAL CUTTING WITH AN ELECTRIC PUNCHER. Notes from illustrated lecture, J. L. Wagner. Gift of Pneumelectric Machine Company, Syracuse, N. Y.

COLLEGE ENGINEERING NOTEBOOK FOR CLASSES IN TECHNICAL SCHOOLS AND COLLEGES, R. E. Moritz. *Boston, Ginn & Company, 1912.*

This notebook is designed for the use of students in civil, mechanical and electrical engineering. It contains 90 sheets of high-grade rectangular-coördinate paper, five sheets of polar-coördinate paper, and five sheets of logarithmic-coördinate paper. The reverse side of each sheet is ruled horizontally with vertical cross-lines at intervals of 2½ cm.

Additional features of the book are lists of the most important mathematical formulæ from algebra, geometry, trigonometry, analytics, and calculus. These are followed by separate pages of carefully selected formulæ from mechanics, surveying, strength of materials, mechanism, machine design, thermodynamics, electricity and magnetism. The back part of the book contains a four-place table of logarithms, and short tables of natural logarithms, trigonometric functions, exponential functions, squares and square roots, cubes and cube roots, reciprocals, and hyperbolic functions. Finally it contains eight sets of type curves.

DIRECTORY OF ENGINEERS AND POWER PLANTS OF GREATER NEW YORK, 1910. *New York, 1910.*

ELECTRICAL INJURIES, THEIR CAUSATION, PREVENTION AND TREATMENT, Charles A. Lauffer. *New York, John Wiley & Sons, 1912.*

The author is medical director of the relief department in the Westinghouse Electric Company at Pittsburgh. The volume is of convenient size for the pocket. In addition to the matter on treatment of electrical injuries, which is concise, the author has an interesting chapter on the effect of occupations on health. Altogether a very interesting little volume.

ENGINEERING SOCIETY OF WISCONSIN. Proc. 1-2. 1909-1910. Gift of Engineering Society of Wisconsin.

FINANCIAL SIGNIFICANCE TO THE GAS INDUSTRY OF THE MODERN BY-PRODUCT OVEN, Paul Schlicht. Read before the Institution of Gas Engineers, June 1907. Westminster. Gift of the author.

GENERAL CONTRACTORS ASSOCIATION. Bull. vol. 1, 1910. *New York, 1910.*

GESCHICHTE DES VEREINES DEUTSCHER INGENIEURE. Nach hinterlassenen Papieren von Th. Peters. *Berlin, 1912.* Gift of the Society.

GOOD ROADS YEARBOOK, 1912. *Washington, 1912.*

HANDBUCH DER BAUSTATIK, A. F. Zschetzsche. *Dusseldorf, 1912.*

HISTORY OF NEW YORK SHIP YARDS, J. H. Morrison. *New York, 1909.*

JAHRBUCH DER LUFTFAHRT, 1912. vol. 2. *Munchen, 1912.*

THE LOCOMOTIVE. vol. 28. Hartford, 1910-1911. Gift of Professor Hutton.

MARINE STEAM TURBINES, G. Bauer and O. Lasche. Translated from the German by M. G. S. Swallow. *New York, 1911.*

MECHANISCHE TRIEBWERKE UND BREMSEN, Dr. St. Löffler. *Munich, R. Oldenbourg, 1912.*

This small work treats of power transmission from the mathematical standpoint, developing formulæ for belts, gearing, rope drive and brakes. It should be of interest particularly to the automobile engineer.

THE MNEMONIC SYMBOLIZING OF STORES UNDER SCIENTIFIC MANAGEMENT, Howard G. Benedict. *1912.* Gift of the author.

MODERNE AUTOMOBIL SEIN BAU UND BETRIEB SEINE PFLEGE UND REPARATUREN, M. Peter. *Berlin, 1912.*

NEW YORK CITY BOARD OF WATER SUPPLY. Contract 101. *New York, 1912.*

NEW YORK STATE ENGINEER AND SURVEYOR. Annual Report 1910, 1911, vol. 1. *Albany, 1911-1912.* Gift of State Engineer and Surveyor.

NOTES ON HEATING AND VENTILATION, J. R. Allen. ed. 3. *Chicago, 1911.*

PRINCIPES D'ORGANISATION SCIENTIFIQUE DES USINES, F. W. Taylor. *Paris.* Gift of the author.

DIE SAUGGASANLAGEN IHRE ENTWICKLUNG, BAUART, WARTUNG UND PRÜFUNG, G. Lieckfeld. *Munchen, 1909.*

STOHMANNS HANDBUCH DER ZUCKERFABRIKATION, A. Schander. ed. 5. *Berlin, 1912.*

WORKMEN'S COMPENSATION FOR ACCIDENTS, F. E. Law. *New York, 1912.* Gift of Fidelity and Casualty Company of New York.

UNITED ENGINEERING SOCIETY

EARTHWORK DIAGRAMS, C. A. James. *White Haven, Pa., 1912.* Gift of the author.

HOW TO PLAN A LIBRARY BUILDING FOR LIBRARY WORK, C. C. Soule. *Boston, 1912.*

NEBRASKA STATE RAILWAY COMMISSION. 4th Annual Report to the Governor, 1911. *Omaha, 1911.* Gift of the commission.

REPORT ON REVALUATION OF RAILROADS AND CANALS, NEW JERSEY, 1911. *Trenton, 1912.* Gift of Chas. Hansel.

SVENSKT PORTRÄTTGALLERI ANDRA SERIEN. I. Svenska Teknologföreningen 1911. *Stockholm, 1911.* Gift of Svenska Teknologföreningen.

GIFT OF WM. PAUL GERHARD

LINDLEY, W H. Anlagen zu dem generellen Projekt der Kanalisation der Stadt Würzburg.

———Barmen-Elberfeld Entwässerung. Denkschrift über die Klärung der Abwässer, vom 14 September 1901.

——— ———Erläuterungs-Bericht zum Projekt einer Klärungsanlage für die Abwässer von Barmen-Elberfeld auf dem Gute Buchenhofen. 1902

——— ———Reise-Bericht der zum studium der Klärung der Abwässer nach England entsandten Commission Elberfeld. 1902.

———Entwässerung der Stadt Homburg v. d. Höhe. *Frankfurt a. M., 1886.*

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CHICAGO PNEUMATIC TOOL CO., *Chicago, Ill.* The anatomy of the Little Giant Commercial Car, 24 pp.

GREENE, TWEED & CO., *New York.* Latest improved Rochester automatic Lubricators, 48 pp.

HART & COOLEY CO., *New Britain, Conn.* Wrought steel ventilators and registers, 48 pp.

HESS-BRIGHT MFG. CO., *Philadelphia, Pa.* Ball bearing hangers, 32 pp.

INGERSOLL-RAND CO., *New York.* "Radialaxe" air driven coal cutters, 20 pp.

MANHATTAN DRILLING CO., *New York.* Dobbins core drills, 20 pp.

NELSON VALVE CO., *Philadelphia, Pa.* Nelson steel valves and fittings, 62 pp.

PNEUMELECTRIC MACHINE CO., *Syracuse, N. Y.* Electric rock drills, 7 pp.

W. B. SCAIFE & SONS, *Pittsburgh, Pa.* Range boilers and high pressure tanks, 35 pp.; Scaife water filters, 32 pp.

SPRINGFIELD BOILER & MFG. CO., *Springfield, Ill.* Boilers and boiler supply material, 47 pp.

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